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Use of Seawater for Air Conditioning

at Waikiki Convention Center

by Mike Williams

submitted to partially fulfill

the requirements of M.S. in Ocean Engineering

University of Hawaii at Manoa

Spring 1994

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Synopsis

A large part of operating costs of a hotel in Hawaii is the cost of energy for air conditioning. Buildings can be constructed to use energy more efficiently by using many methods, however, some of these methods conflict with other concerns, aesthetics for example. Thus the process of designing and building an energy efficient hotel often involves trade-offs between energy efficiency and other objectives.

The method proposed herein to reduce energy costs is to introduce seawater, pumped from the deep ocean at a temperature of approximately six degrees celsius, directly to heat exchangers which cool the chilled water circulating in the building air conditioning system. The energy required to run the system would be reduced to only the cost of the seawater pumps, the fans and controls. The savings would be in the operating costs of the seawater pumps versus the cost to the compressors of a conventional air conditioning system.

One project which could conceivably employ such a system is the proposed future convention center in Waikiki. This paper presents a site specific conceptual design based on the proposed location of that project.

The proposed system would employ a high density polyethylene pipe of more than a meter diameter to deliver enough cold water to the site to accommodate the air conditioning load. The pipeline would have to be approximately 7.2 kilometers long and extend 5.6 kilometers offshore. The pipeline can be built with current technology.

Similar pipelines are operating at Keahole Point on the island of Hawaii, but this design would be the longest attempted to date. Current technology also exists to construct heat exchangers of either titanium or zinc-clad aluminum. It is likely that these could be ordered "off the shelf". Bio-fouling would not be a problem for either the pipeline or the heat exchangers as long as the cold water is not exposed to sunlight.

A seawater air conditioning system would be expensive to construct. However, the initial capital cost would be offset by the savings in electricity costs over the course of the first seven to nine years. Over the life of the system, the savings would be well into the tens of millions of dollars.

Table of Contents

Introduction

- A. General Introduction
- B. History of Seawater Air Conditioning

Project Background

- A. Description of Convention Center Project
- B. Description of Project Site

Requirements for Building

- A. Cooling Requirements
- B. Cold Water Requirements

Cold Water Pipe Design

- A. Material
- B. Pipe Length
- C. Pipe Diameter
- D. Pipe Flow Characteristics
- E. Pipe Wall Thickness
- F. Temperature Gains
- G. Calculations of Cold Water Pumping Power Requirement
- H. Effects Of Waves And Currents
 - 1. Description of Forces
 - 2. Force Analysis of Various Configurations
 - 3. Scour
 - 4. Vibrations
 - 5. Discussion of Combined Effects/Pipe Design
- I. Environmental Considerations
- J. Fouling
- K. Alternative Design: Directional Drilling

System Design

- A. Seawater Heat Exchangers
 - 1. Materials
 - 2. Bio-fouling
 - 3. Design
- B. Pumps
- C. Heat Gains

Economic Analysis

- A. General Approach
- B. Estimate of Capital Costs
- C. Operating Costs
- D. Energy Analysis
- E. Cash Flow Analysis

Summary and Conclusions

Appendices

Appendix A - Sample Calculations and Results

- A-1 Pipe Flow Calculations
- A-2 Pressure Change Calculations For Cold Water Pipe With
Calculation of Maximum Allowable Negative Pressure
- A-3 Temperature Gain Calculation For Cold Water Pipe
- A-4 Hydrostatic Forces (External) On Cold Water Pipe

Appendix B - Engineering Properties

- B-1 Relative Roughness
- B-2 Friction Factors
- B-3 Specific Heats of Seawater
- B-4 Mechanical Properties of Plastics
- B-5 Mechanical Properties of Various Materials

References

List of Figures

- 1 - Seawater Air Conditioning System
- 2 - Map of Waikiki with Three Proposed Convention Center Sites
- 3 - USGS Bathymetry Map
- 4 - Bathymetry Along Pipeline Route
- 5 - Architects Hawaii Ltd.'s Rendition of Convention Center Site
- 6 - Seawater Flow
- 7 - Temperature Rise
- 8 - Typical Seawater Temperature Profile
- 9 - USGS Bathymetry Map With Pipe Route
- 10 - Pipe Flow Model
- 11 - Current and Wave Direction
- 12 - Configuration I, Pipeline Flat on Bottom
- 13 - Free Body Diagram
- 14 - Configuration II, Pipeline Partially Buried
- 15 - Configuration III, Pipeline Completely Buried
- 16 - Configuration IV, Pipeline Supported Above Seafloor
- 17 - Configuration V, Catenary Mooring of Pipeline
- 18 - Lee-Wake Effect
- 19 - Equilibrium Scour Profiles
- 20 - Log Mean Temperature Difference

List of Tables

- 1 - Convention Center Size Estimate
- 2 - Convention Center Air Conditioning Load Projection
- 3 - Comparison of Pipe Sizes and Thicknesses
- 4 - Heat Exchanger Design Criteria
- 5 - Capital Costs of Seawater Air Conditioning System

Introduction

GENERAL INTRODUCTION

Air conditioning is very important to buildings in Hawaii. Hotels, shopping malls, restaurants and office buildings all rely on air conditioning to keep their guests, customers, patrons and workers comfortable. This is particularly important in the tourist industry where hotel operators want to keep their guests happy so that they might recommend Hawaii to their friends or return themselves.

Because of the importance of air conditioning, Hawaii business owners are willing to pay a high price to air condition their facilities, and they do. Air conditioning is expensive to operate, further driving up the costs of goods and services in Hawaii. Most of the operating costs are associated with the tremendous power consumption of air conditioning systems. About 1 kW of electricity is required to provide 1 ton of air conditioning in a conventional air conditioning system. The cost of electricity in Hawaii is high; on Oahu the cost is about \$.10/kWh. There are well over 20,000 hotel rooms in Waikiki alone, each requiring approximately 1 ton capacity of air conditioning. This could amount to over \$48,000 per day just for hotel room air conditioning.

Alternate technologies are available to help reduce the cost of air conditioning. The natural wind patterns could be used in some applications if buildings are designed to take advantage of them. However, this is not too attractive to hotels, for example, where

visitors expect reliable climate control. It is also impractical to retrofit existing buildings for natural wind usage.

Another way to air condition buildings without consuming a large amount of electrical power is through the use of the naturally cold water which can be pumped up from the deep ocean. Such a system does not need large, power consuming condensers, and would not use chlorofluorocarbon products which are blamed, in part, for the depletion of the ozone layer. The few buildings using the proposed deep seawater systems operate their air conditioning for approximately 20% of the cost of conventional systems.

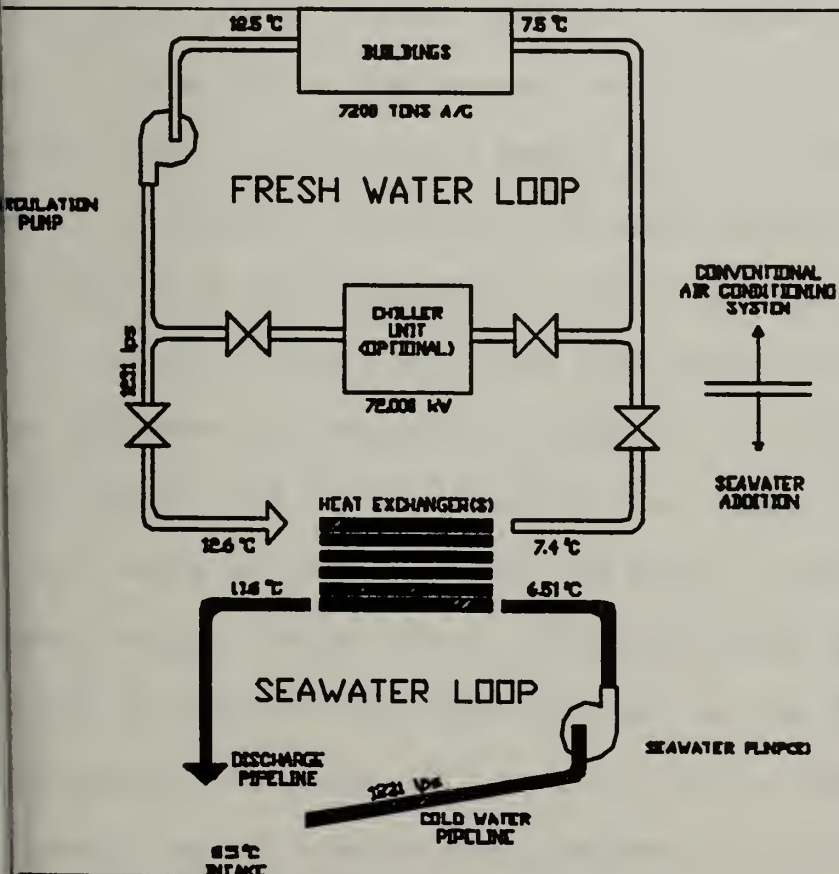


Fig. 1 - Seawater Air Conditioning System
(adapted from Van Ryzin, 1991)

The idea behind such a system is that cold seawater pumped from the deep ocean is used to cool the water circulating in the air conditioning system through the use of heat exchangers (see Figure (1)). The heat exchange between the two fluids replaces the operation of the condensers or chillers in a conventional air conditioning

system. This process could be used to completely replace the condensers, or could be used in conjunction with the condensers if either the seawater is not cold enough to provide 100% of the cooling required at peak times, or if a back-up system to the seawater system is desired. This paper is intended to present a site specific conceptual design for Waikiki, Hawaii.

HISTORY OF SEAWATER AIR CONDITIONING

The idea of using natural sources of cold water for air conditioning is not new. It has been appearing in literature for over sixty years (Ciani 1980). Over the years, a variety of studies have shown that energy savings from use of seawater or lake water can result in from 70% to 90% energy savings in places where there is a large, highly concentrated demand for air conditioning. The rows of hotels in tropical sites such as Miami Beach and Honolulu are sited studies as good candidates (Ciani 1980; Hirshman and Kirklin 1977). Less obvious sites have also been studied by the Naval Facilities Engineering Command. In addition to sites such as Naval Shipyard Pearl Harbor and Pacific Missile Test Center in Point Mugu, CA, sites as far north as Cutler, Maine and Naval Security Group Activity in Winter Harbor, Maine (Ciani 1980) have also been looked at for feasibility of seawater air conditioning. In the case of Winter Harbor, a preliminary design in 1977 resulted in the conclusion that modifying the existing air conditioning system to utilize seawater was the most economical alternative to more conventional upgrades.

In all the Navy investigated 241 sites in the 1970's for potentially economic use of seawater for air conditioning. At least twenty-five of these sites were identified as having potential. Ten of those were identified as having a high potential. The sum of the energy savings at these ten sites was estimated at 23,000 MWh, at a cost at that time of \$1.3 M per year (Ciani 1980).

A 1977 study by the Energy Research and Development Administration, ERDA, included a preliminary design of a large Miami Beach seawater air conditioning system. That study estimated that a 20,000 ton system would cost from \$20.4 to \$23.4 M, and would include a 2,000 foot (6.7 km) cold water supply pipeline (Roach, 1976). The cost estimates cited were admittedly crude at the time and these studies were based on 1977 prices. Similar construction would probably cost considerably more in 1994 dollars, however, construction materials and techniques for the costliest item, the cold water pipeline, have changed over the years. Many of the previous studies were based on steel pipe, whereas more recent pipeline installations have used polyethylene pipe. Despite the newer technologies, the cold water supply pipes still represent the largest portion of the capital costs of a seawater air conditioning system. The same study also concluded that cold water from fresh water bodies such as the Great Lakes, other large lakes and ground water from the northern third of the United States are also good sources for air conditioning systems (Birshman, Kirklin 1977).

In recent years, the idea of using seawater for air conditioning has gone beyond the study and preliminary design stage. One system currently using seawater to cool buildings is in Nova Scotia, Canada. In this system, water is pumped out of Halifax Harbor. Although Nova Scotia does not have the cooling requirements of Hawaii, the owners of that complex are very satisfied with the performance and speed at which the capital investment is being paid off; they plan to cool future expansions of their facility in the same manner (Tenbruggemcate 1993).

At the Natural Energy Laboratory of Hawaii, another such system is also currently in operation, and cools the two main buildings at that site. This system uses cold ocean water from an approximate depth of 600 meters, supplied by means of a high density polyethylene pipe.

The Natural Energy Laboratory of Hawaii (NELH) is located at Keahe Point on the west coast of the Island of Hawaii. There, a 30 cm diameter deep seawater pipeline has been in operation since 1982. Since 1987, four other pipelines have been added at the same location. Two 38 cm pipelines were deployed by a private firm. The state of Hawaii deployed another 45 cm pipeline for "back-up" and a 1 m diameter pipeline now supplies cold water to both the State of Hawaii Ocean Science and Technology Park and the Ocean Thermal Energy Conversion (OTEC) research project at NELH, which is sponsored in part by the U.S. Department of Energy (DOE). The pipelines supply cold water for a variety of purposes, including aquaculture, OTEC research, pro-

ction of fresh water and the air conditioning of the laboratory
buildings (Daniel 1989).

All of the pipelines mentioned above are constructed of high density polyethylene. A variety of design options have been used to successfully deploy and anchor these pipelines. The techniques which have proven successful there will be useful in future projects.

In this conceptual design, the operating temperature of the cold water can be assumed to be similar to those of the system which has been successful off Keahole Point, Hawaii. The load on the air conditioning system is based on a rough estimate of the air conditioning load of a future convention center in Waikiki, which has yet to be designed. The amount of water supplied will be based on that load. The seawater will cool the fresh "chilled" water, which will circulate in the air conditioning system. The latent heat is included in the gross estimate of the air conditioning load at this stage. Future design phases would have to consider the system in much more detail.

Background

DESCRIPTION OF CONVENTION CENTER PROJECT

This conceptual design is intended to be site specific. The site chosen is one of three possible sites for a future convention center project in Waikiki. Currently, the convention center project is only in the preliminary stages of planning. There were three sites in

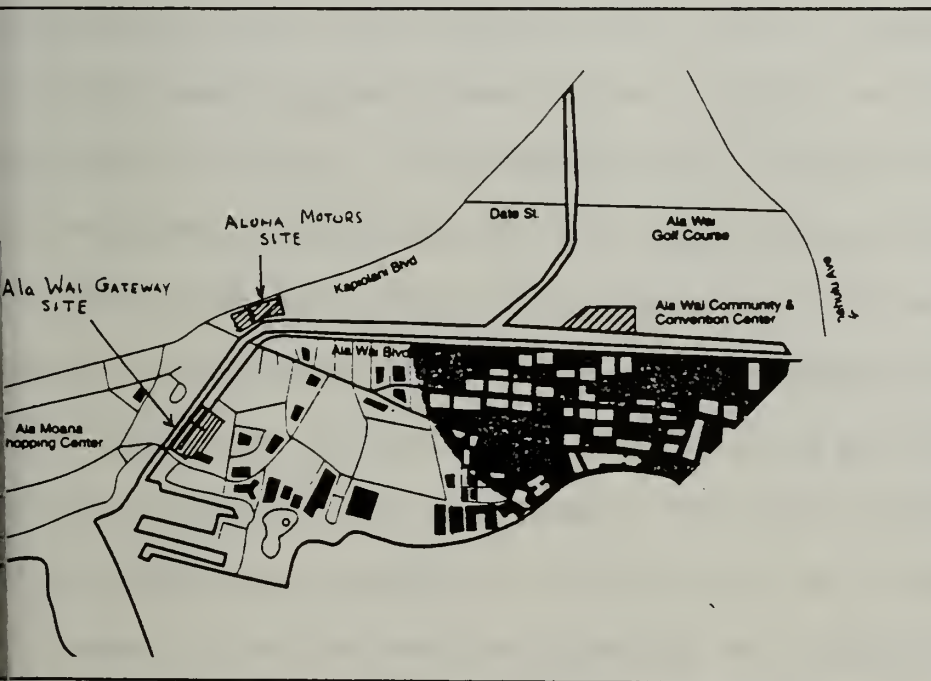


Fig. 2 - Map of Waikiki with Three Proposed Convention Center Sites

Waikiki which were under consideration (see Figure (2)). One of the sites is at the old Aloha Motors site, the second is on the Ewa end of the Ala Wai Golf Course, and the third is across the Ala Wai from the Aloha Mo-

rs site, closer to the Ala Wai entrance. Design details are nonexistent. Nevertheless, enough information is known to present a conceptual design for an air conditioning system which would use seawater for cooling.

As might be expected, the larger the expected air conditioning load, the easier it is to justify the capital expense of constructing a cold water pipe from Waikiki to water depths capable of use in an air conditioning system. For that reason, the convention center proposal containing the greatest need for a larger air conditioning system was selected for the basis of this concept design. That is the proposal to build a convention center and hotel complex on the old Aloha Motors site, as originally put forth by Sukamto Holding Corp. The other two proposed plans did not include any hotel rooms, and were smaller overall. The proposed hotel complex for the Aloha Motors site originally consisted of up to four towers, each up to four hundred feet high, with 2800 hotel rooms, but this has since been changed. In December of 1993, the state agreed to purchase the Aloha Motors site from Sukamto, and is expected to put the project out for separate proposals for development. This will probably result in a scaled down hotel portion of the facility (an estimate of only 500 hotel rooms is now included) and possibly an expanded or expandable convention facility.

DESCRIPTION OF PROJECT SITE

A design of the seawater air conditioning system for this project requires a cold water supply pipe of large enough diameter to deliver an adequate cold seawater supply at a temperature low enough to meet the cooling requirements of the convention center complex. Water at the required temperature, approximately 6°C, is generally found at

depths of 500 meters and greater. Such a pipe could, in concept, be laid in the Ala Wai Canal, adjacent to the Aloha Motors site and out through the Ala Wai Boat Harbor toward deeper water. There is a dredged channel in front of the Hilton Hawaiian Village, but it may not be practical to use it for a cold water pipe of the size and route required.

Although, the water surrounding the Hawaiian islands is very deep, 1000 meters and more in places, as can be seen from the bathymetry shown in Figure (3) and summarized in Figure (4), the ocean floor off Waikiki slopes gradually for the first few kilometers. This gentle slope makes it necessary to construct a cold water supply pipe much longer and more expensive than those constructed at Keahole Point. The area of the pipeline exposed to the surf zone would also be larger than that of Keahole Point, increasing the difficulty associated with burial or securing the pipe to the bottom. This would also increase the cost. These things must all be considered in the design and in the economic analysis of such a project.

Mercator Projection

Contour Interval - 10 meters corrected
for the speed of sound in seawater.

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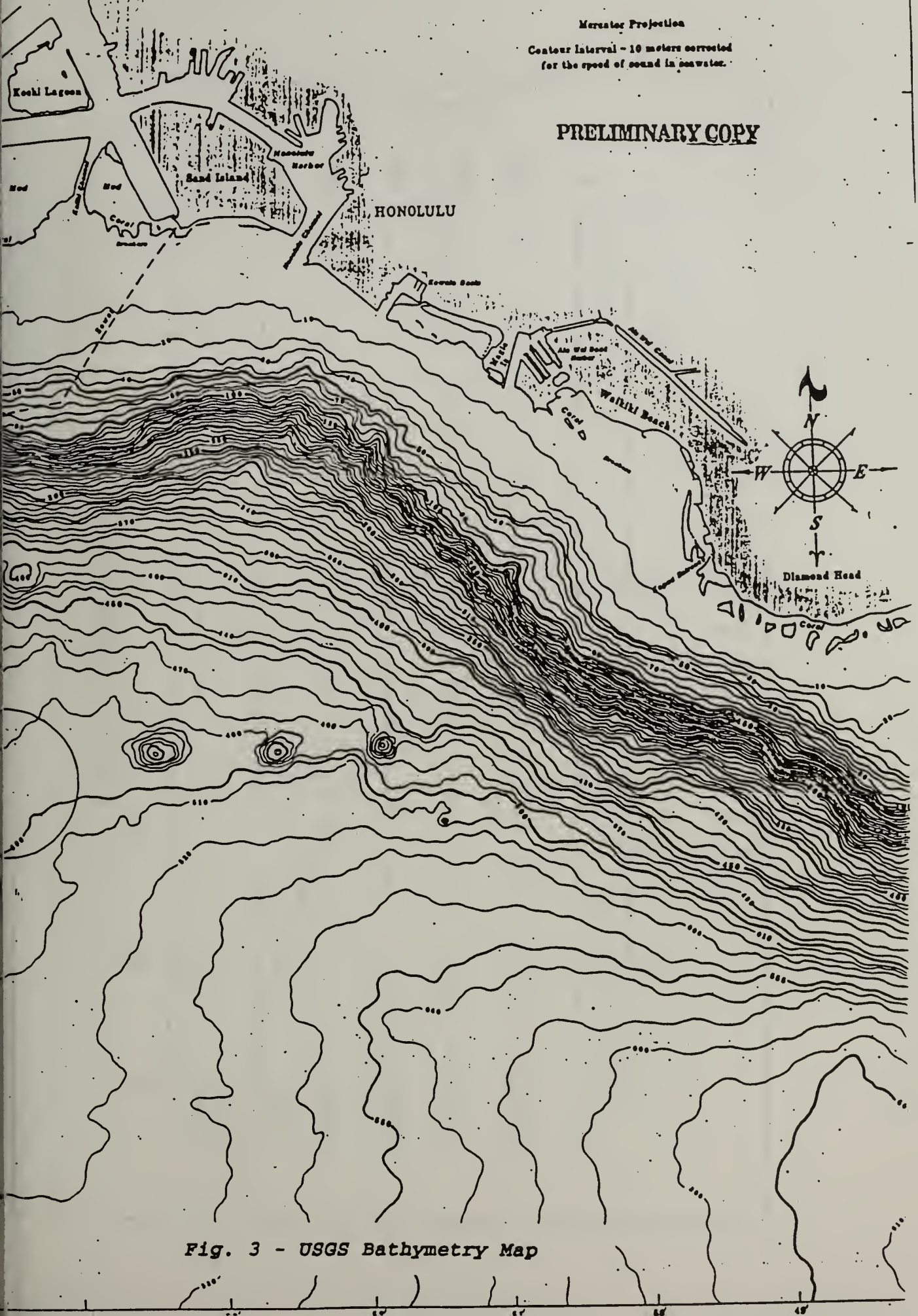
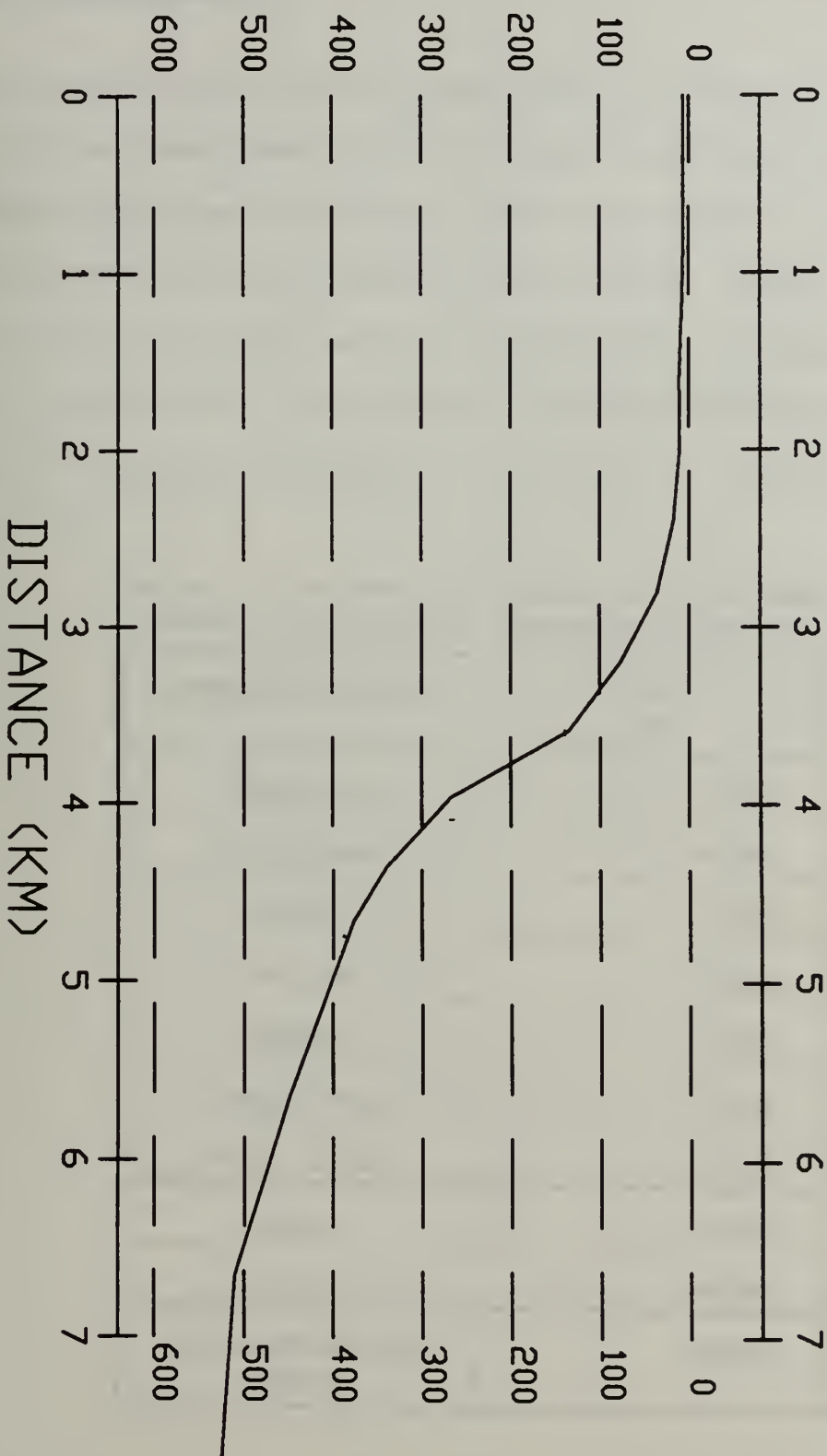


Fig. 3 - USGS Bathymetry Map

BATHYMETRY

DEPTH (METERS)



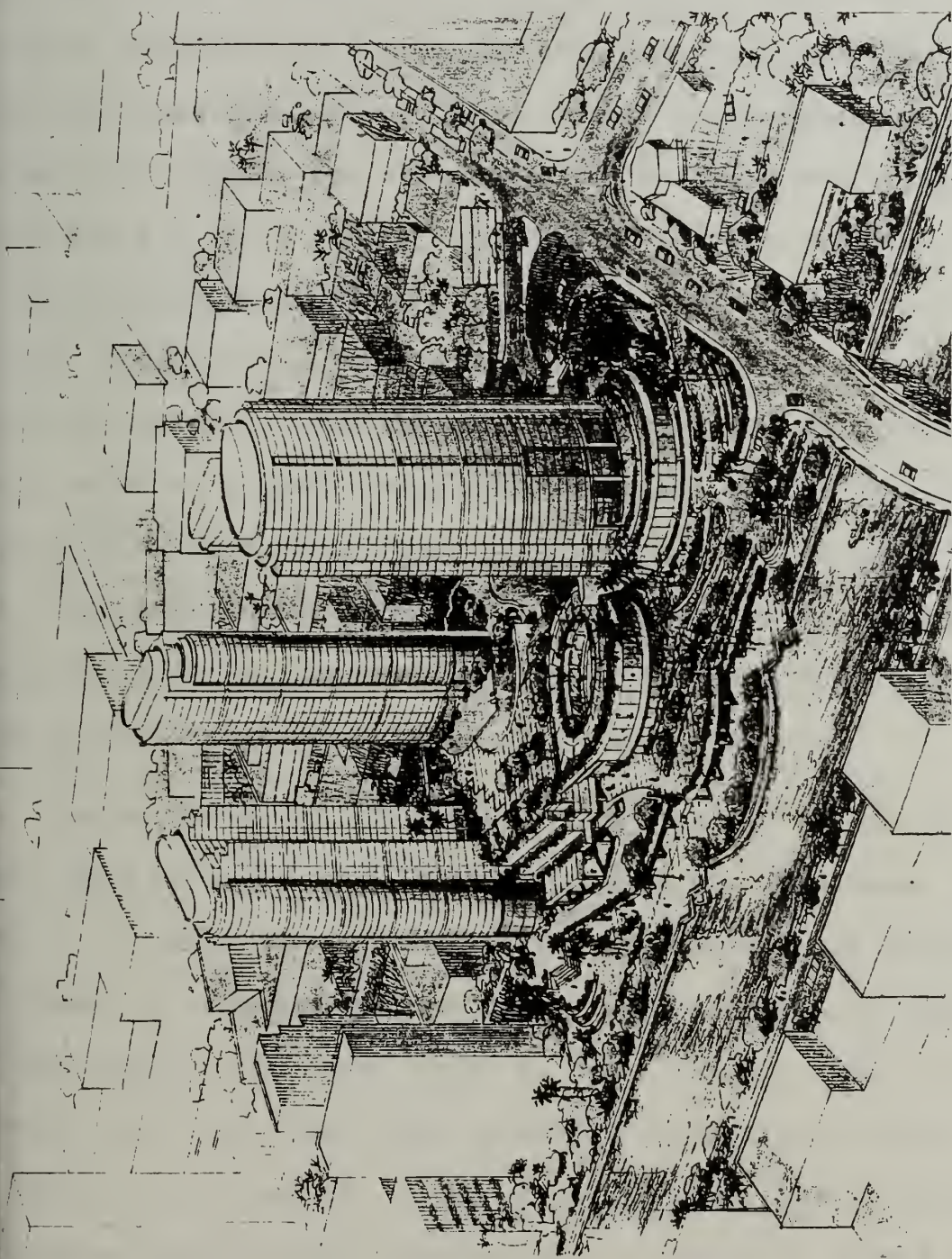
Requirements for Building

COOLING REQUIREMENTS

The proposed Aloha Motors development is shown in Figure (5). A summary of the significant building characteristics is shown in Table (1). Judging from the information available, however, this appears likely to include the first phase of the proposed complex. To comply with state and city requirements, the complex is intended to have the capacity to expand in a second phase, increasing convention space to 200,000 s.f., meeting room space to 115,000 s.f. and ballroom space to 65,000 s.f.

*Table (1) Convention Center Size Estimate
taken from Sukanto Executive Summary*

CONVENTION CENTER	GSF (ft ²)
Exhibit Halls	200,000
Meeting Rooms	100,000
Ballroom	35,000
Pre-Function	75,000
Registration	25,000
Offices, Toilets	40,000
Service, Support	200,000
Sub Total	675,000
Parking	280,000
TOTAL CONVENTION CENTER	955,000



ARCHITECTS
HAWAII
LIMITED
STRONG, TUSHER
AND ASSOC

FEBRUARY 1 1983



HAWAII CONVENTION CENTER

Fig. 5

Also not included in Table (1) is the space for hotel rooms. The total proposal originally included four towers, each up to 400 feet high (40 stories) for a total of 2800 hotel rooms. This was a distinguishing characteristic from the other two proposals being considered from the stand point of air conditioning load. Table (2) takes those things into account and summarizes the building characteristics on which the air conditioning load estimates for a seawater air conditioning system could be based. The first two estimates in Table (2) reflect the original Sukamto proposal. As stated earlier, however, it can be anticipated that the complex to be built may not have as many hotel rooms as Sukamto originally proposed. To that end, a third estimate of air conditioning load is shown in Table (2) with only 500 hotel rooms and thus 500 tons of air conditioning projected. The other characteristics will probably be about the same since they comply with the criteria set forth by the State.

The estimates of air conditioning load can get fairly complicated when building details are available. In this case, the building details are sketchy at best, and the estimate of air conditioning requirement is, therefore, admittedly rough. A "rule of thumb" that can be employed for the hotel rooms is from .75 to 1.0 ton of air conditioning per hotel room. When referring to cooling capacity, one ton of capacity is equal to 3.5 kW or 200 BTU/min, and is defined as the continuous cooling rate provided over a period of twenty-four hours in melting one short ton (906 kg) of ice. In the third estimate, one

n per room was used for simplicity, since the 500 room capacity is
guess at best.

Table (2) Convention Center Air Conditioning Load Projections

CONVENTION CENTER	GSF (ft ²)	1ST ESTIMATE (1st rough guess) (tons)	2ND ESTIMATE (comparison to Blaisdell) (tons)	3RD ESTIMATE (combined after state land purchase) (tons)
Exhibit Halls	460,000		3382	3382
Meeting Rooms	115,000		1008	1008
Ballroom	65,000		448	448
Pre-Function	75,000		551	551
Registration	25,000		95	95
Offices, Toilets	40,000		151	151
Service, Support	200,000		1500	1500
Sub Total	980,000	3920	5634-7134	5634-7134
Hotel Space	700,000	2100-2800	2100-2800	500
Parking	280,000	0	0	0
Total Convention Center	1,960,000	6020-6720	7834-9934	6134-7634

The other areas of the complex are not covered by any such "rule
thumb." The loads and ventilation requirements in the different
areas vary widely. Exhibition halls full of people require a high ca-
pacity for air conditioning per square foot, as do restaurants and
kitchen areas. Ballrooms may also be either full of people or vacant.
Initial estimate can be based on a gross assumption that the over-
requirement of 980,000 square feet requires approximately the
amount of air conditioning per square foot as the hotel room.
This is the methodology applied in the first estimate in Table (2). A

total load of 6720 tons can then be derived. A comparison to the other local convention center is made in the second estimate of Table). In this estimate, the sizes of the chillers for the various areas of the Blaisdell Exhibition Center, a much smaller facility, were used. A direct per square foot relationship is assumed. The purpose of some of the areas described in the Sukanto proposal is unclear, so an assumption was made for the service areas that they may be either heavy for that area, 1500 tons, or not air conditioned at all, such as a loading dock area. Using this comparison, the expected load for the entire complex may be between 7834 and 9934 tons. The comparison to the Blaisdell Exhibition Center indicates that the first gross estimate may be a little low, but is at least of the same order of magnitude.

Some additional factors may shed some light on this rough estimate process, one being that the Blaisdell complex was built many years ago when little attention was paid to energy conservation. Construction technology has improved in that respect and a new convention center design would hopefully be more energy efficient.

Another factor is that the larger facility can be expected to be easier to air condition due to less wall area exposed to the elements per square foot of building. This may be a big factor. For example, a 400,000 s.f. building would have 2524 linear feet of wall (if it were a square). But, by comparison, a building with 1/10 the area, 40,000 s.f., would still have 800 linear feet of wall, 1/3 the amount of the larger building. Of course, ratios of roof areas would theo-

etically be the same as the building square footage ratios, but the efficiency of the larger building is apparent.

Yet another factor is that it is unlikely that the ballrooms, exhibition spaces, hotel rooms and restaurants would all be cooled at once. It could be argued, therefore, that a conservative utilization factor of .75 should be used to multiply the total possible load in the second estimate, significantly reducing the expected load. This is less of a consideration in the third estimate since fewer hotel rooms would indicate that the facility would be used more purely as a convention facility only. Taking the factors mentioned above into account, a refined version of the estimates is shown as the third estimate. The load used in the cold water requirements calculations is 7200 tons. This is on the higher, more conservative side of the estimate between 6134 and 7634 tons.

COLD WATER REQUIREMENTS

With an air conditioning load of 7200 tons, enough cold water must be supplied to the heat exchangers to remove 25,200 kW of thermal energy from the fresh water which circulates in the air conditioning system. By definition, 1 ton equals 3.5 kW of thermal energy. From the experience of previous projects, a 5°C increase of the seawater temperature in the heat exchangers can be assumed. Then the following relation must be true,

$$C_p \cdot 5^{\circ}\text{C} \cdot M = 25,200 \text{ kW} \quad \text{Eq. (1)}$$

where:

M = mass of seawater through heat exchanger

C_p = specific heat of seawater $\cdot \frac{\text{kJ}}{\text{kg } ^\circ\text{C}}$

$$= 3985 \frac{\text{J}}{\text{kg } ^\circ\text{K}}$$

(See Appendix B-3 for specific heats of seawater at various temperatures and salinities.)

$$3985 \frac{\text{J}}{\text{kg } ^\circ\text{C}} \cdot 5^\circ\text{C} \cdot M \left(\frac{\text{kg}}{\text{s}} \right) = 25,200 \text{ kW} \left(\frac{\text{kJ}}{\text{s}} \right) \quad \text{Eq. (2)}$$

Solving for M :

$$M = 1265 \frac{\text{kg}}{\text{s}}$$

$$\text{Volumetric Flow } Q = \frac{M}{\rho} = \frac{1.265 \frac{\text{kg}}{\text{s}}}{1027.6 \frac{\text{kg}}{\text{m}^3}} = 1.231 \frac{\text{m}^3}{\text{s}} \quad \text{Eq. (3)}$$

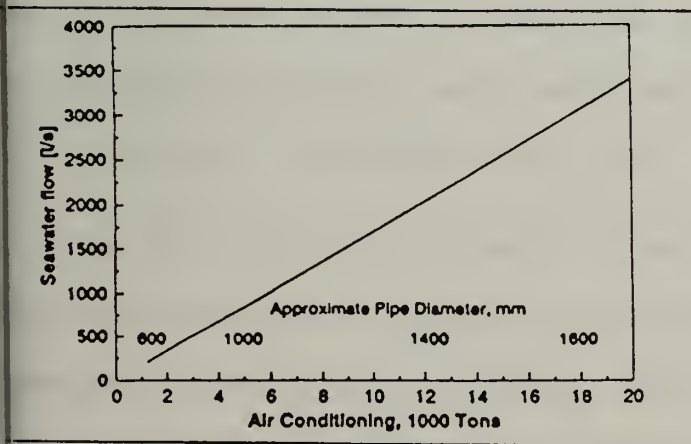


Fig. 6 - Seawater Flow
From Van Ryzin, 1991)

The flow must then be 1231 lps (19,510 gpm). When specifying a heat exchanger, some vendors are more receptive to gallons per minute. The 1231 lps value agrees fairly closely with Figure (6) which was previously determined (Van Ryzin 1991).

Knowing the volumetric flow requirement to achieve the desired cooling allows for design of the cold water pipe and seawater-freshwater heat exchanger.

Cold Water Pipe Design

MATERIAL

In many older designs for ocean pipelines the basic pipeline material used in the petroleum industry, for example, is steel. External and internal coatings were specified for corrosion protection and hydraulic flow enhancement. Concrete jackets were specified to provide added weight, thereby providing stability. This weight was considered too great for pulling into place, and the pipeline sections would have had to be voided during the pulling operations. This was an expensive and difficult process. Newer technologies make things easier.

Plastic pipe, now available in large diameters, has several important advantages over steel pipe in this application. It will not corrode in seawater and it has good thermal insulation properties. Two candidate materials are, high density polyethylene pipe (HDPE) and fiberglass reinforced resin pipe (FRP).

FRP has excellent strength-to-weight characteristics. It can withstand burial and external pressure. Its axial strength-to-weight characteristics permit extremely long sections to be bottom-pulled. It is resistant to seawater (minimum life of 40 years), has excellent thermal insulation properties, and is available in specific gravities ranging from 1.6 to 2.08. This is an advantage where negative buoyancy is desired.

The high density polyethylene pipe is attractive because it can, in some situations, be continuously extruded at the site. It is available "off the shelf" in diameters up to 63 inches in 40 foot sections. It has been successfully deployed in the ocean in the following manner. Its end is sealed, and it is extruded into quiet waters. Concrete weights are bolted to the pipe at regular intervals while it is afloat. These concrete anchors, once installed, do not permit bottom pulling. When a sufficiently long section is assembled, it is floated to the site and installed by controlled flooding. This technique is good for protected waters. In the Waikiki design, the Waiakea Wai Canal could be the staging area.

Polyethylene, a type of plastic, is a polymer made up of long chain molecules. Polymers can be formed in nature or can be manufactured by man from natural materials. Polyethylene falls into the category of plastics known as thermoplastics. These plastics soften when heated and can be reformed many times. Pipe made from polyethylene can be extruded in various lengths. Other parts such as fittings can be formed by injection molding. Other thermoplastics include polyvinyl chloride (PVC), polypropylene (PP) and acrylonitrile butadiene styrene.

Polyethylene's characteristics make it a good choice for a cold water supply pipeline for a number of reasons. Its resistance to corrosion is excellent, giving it a great advantage over steel. It has adequate stiffness and retains its cross section, yet it is very light compared to steel, making it easier and cheaper to handle at

e site. It also is a better insulator than steel. This is an advantage in areas where the cold water pipe is routed through shallower, warmer water.

There are, however, drawbacks of HDPE. One major disadvantage which arises from polyethylene's lightness is that with a density of 0.960 to 0.965 g/cm³, the pipe is buoyant even when flooded. It therefore requires a great deal of effort to anchor the pipe in place. This must be addressed in the design. Another consideration is that, as with many plastics, the creep modulus of HDPE is important. This can affect the service life of the pipe if not factored into the strength calculations (Rubin 1990). Another drawback is the susceptibility to abrasion that could occur due to motion of the pipe while in contact with the sea floor.

The advantages of high density polyethylene (HDPE) appear to outweigh the drawbacks, and therefore, it is the material chosen for the cold water pipeline. This choice is made easier mainly due to the successful experiments at Keahole Point.

PIPE LENGTH

The length of the cold water pipe is dependent on the bathymetry offshore of Waikiki and the depth at which the required water temperature can be obtained. The required temperature of the intake water at the heat exchanger is expected to be in the vicinity of 6.5°C, from a rough estimate based on the desired performance of the seawater - freshwater heat exchanger. From Figure (7), a temperature rise in the

ld water supply pipe of
 ss than 0.25°C might be ex-
 cted; however, Figure (7)
 based on a study which em-
 oyed a much shorter (220
 pipeline (Van Ryzin
 91). In light of this, an
 itial estimate of 0.5°C

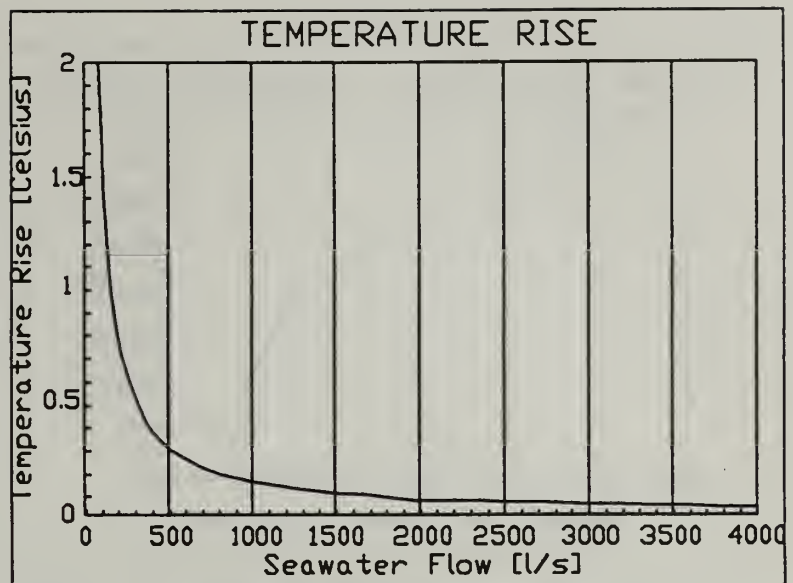
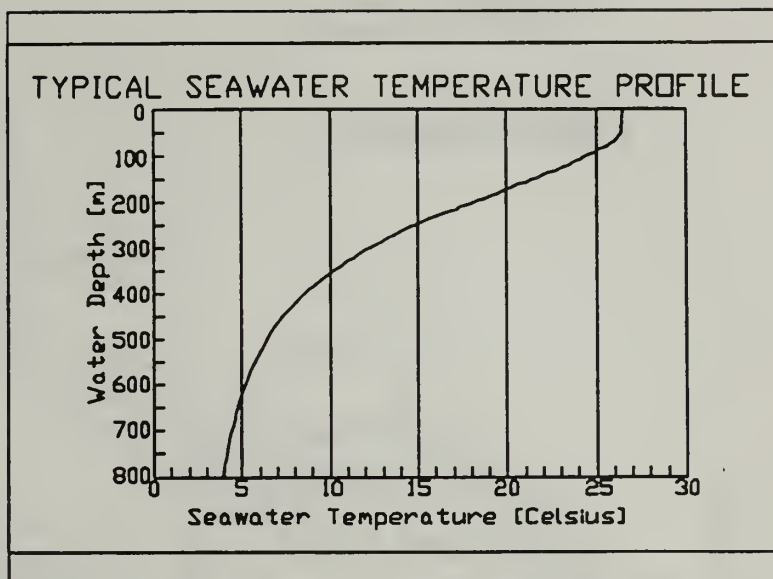


Fig. 7 - Temperature Rise

se in intake water tempera-
 re can be used to establish
 e pipeline length. Once the length is determined, a heat gain calcu-
 tion may be performed for the length of the pipe and if necessary,
 e pipe design length could be adjusted to a length and depth where
 lder or warmer water could be obtained, whichever the case may dic-
 ce. This iterative process could be performed several times to opti-
 ze the design, however, the accuracy of the known conditions would
 t warrant more than one or two iterations at this stage. A calcula-
 on of heat gain is made to verify the reasonability of the 0.5°C
 temperature gain assumption in Appendix A-3. This calculation pre-
 sents a temperature gain of only 0.1°C , but 0.5°C will be used as a
 uservative estimate.

Since the desired heat exchanger seawater intake temperature is
 6°C and the assumed heat gain in the intake pipe is 0.5°C , the in-
 of the seawater supply pipe should be positioned in a region
 re 6°C seawater will be encountered. Figure (3), a yet unpublished

preliminary U.S. Geological
 Survey (USGS) map, indi-
 cates the bathymetric
 contours as accurately as
 any available source can at
 this time. From a typical
 seawater temperature pro-
 file, Figure (8), a depth
 can be assumed at which 6°C
 water may be obtained. The



**Fig. 8 - Typical Seawater Temperature
 Profile in Hawaii**

depth indicated by Figure (8) is approximately 525 m.

Figure (9) indicates a pipeline route to 525 m depth using the
 USGS map, Figure (3), as a reference. The shortest straight line dis-
 tance to the desired depth is shown to be about 5.5 kilometers from
 the Ala Wai Harbor entrance in a south by southwest direction.

The length of the pipe also needs to be adjusted for the depth
 (vertical component of distance) and any possible zig-zag required
 route since bottom obstacles may be encountered and a straight
 pipeline is not a reasonable expectation. Large rocks, coral
 reefs and natural geological features will inevitably cause small de-
 viations.

The vertical component of 500 meters is added to the horizontal
 component of 5500 meters to get 5522 meters. In light of possible ob-
 stacles and bending of the pipe in suspended or catenary support con-

LEGEND

Mercator Projection
Contour Interval - 10 meters corrected
for the speed of sound in seawater.

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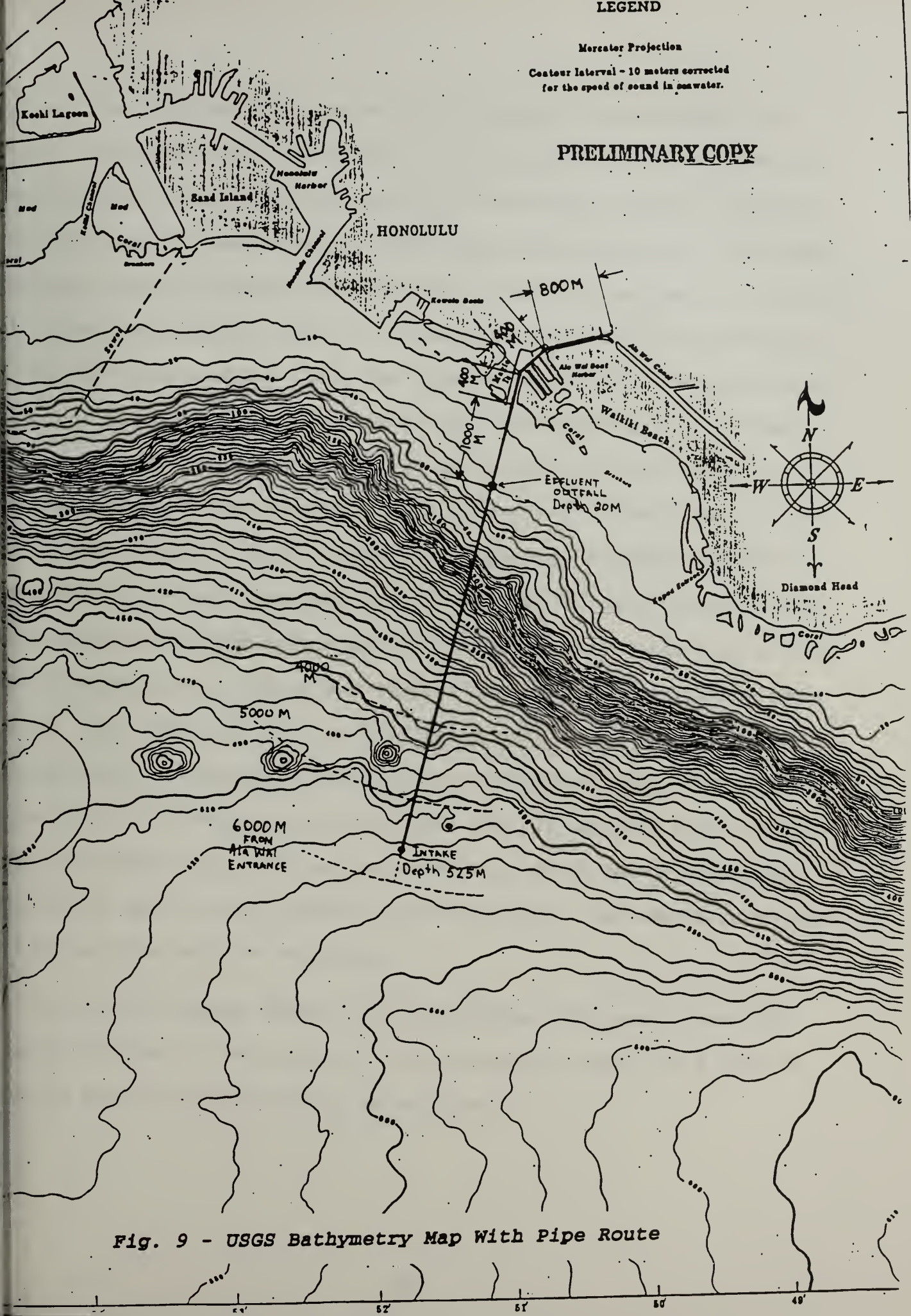


Fig. 9 - USGS Bathymetry Map With Pipe Route

gurations, 5600 meters is used as the distance from Ala Wai entrance. In addition to 5600 meters of pipe required for the distance from the pipe inlet to the entrance of the Ala Wai Harbor, the pipe must continue for approximately 400 meters into the harbor, then bend right and continue another 400 meters to reach the entrance of the canal. After entering the canal, it continues about 600 to 800 meters to the convention center site. The pipe is exposed to a warm water environment throughout these sections. This becomes relevant in the heat gain calculations. The total length of pipe is 7200 meters for this design, more than 3 times the length of any of the pipes at Keahe Point. However, this is a similar distance to that evaluated from the Miami study (Hirshman and Kirklin 1977), with favorable economic results. The reasonableness of extending the intake pipe so far should be considered during any further design processes, since the capital cost involved with the cold water pipe is so high. For the calculations of temperature gain in the cold water pipe, the total distance of 7200 meters can be used for now. It will be shown by calculations herein that the shallow areas are the areas in which the temperature gain is the greatest and even there, the temperature gain inside the pipe will be very small.

For future design phases to be completed, the pipe route indicated by Figure (9) would have to be surveyed in detail and the temperature profile would have to be verified.

PIPE DIAMETER

Since the cold water requirement of 1231 liters/sec has been established, the inside diameter and flow velocity can be determined. Of course, the faster the flow, the smaller diameter pipe can be used and visa versa. Flow velocity also determines the pressure required. A large of a negative pressure imposed on the cold water pipe could collapse it, so care must be taken, after a velocity is picked, to determine the pumping requirements. It is advantageous to use as high a velocity as allowable, since the high velocity saves on pipe cost by reducing the diameter and temperature gain due to losses to the outside seawater. Conversely, a slower velocity requires less pumping energy because of less head loss. The final design should seek an optimum between pumping costs and pipeline costs.

Previous research indicates that a good starting point, as Figure (6) shows, is an 1150 mm diameter pipe. A pipeline of a slightly smaller diameter has been deployed successfully off Keahole Point, Hawaii, and is well within the state of the art (Van Ryzin, 1991). For example, with a diameter of 1150 mm, the calculation of velocity is as follows:

$$\text{Volumetric Flow} = \frac{\pi D^2 V}{4} = 1.3 \frac{\text{m}^3}{\text{s}} \quad (1300 \text{ lps}) \quad \text{Eq. (4)}$$

where:

D = pipe diameter (inside) = 1.15m

V = flow velocity

This calculation yields a flow velocity of 1.25 m/s. With an estimate of inside diameter and velocity determined, the length of the pipe must be determined to establish pressure and temperature losses. This conceptual design will be based on a 1200 mm pipe which has a flow velocity of 1.329 m/s, as explained in Appendix A-1. A more refined selection of pipe size can be made when the flow characteristics are better defined.

PIPE FLOW CHARACTERISTICS

Understanding the characteristics of the seawater flow both inside and outside the cold water pipe is important in determining the pressure drop and head loss as well as the heat transfer characteristics of the system. One objective is to determine the pressure loss, and hence, the power requirements for the intake pumps to achieve the desired flow rate for the inside of the pipe. Some other assumptions are required to make calculations regarding the flow through a pipe. One is that the flow is incompressible so as to satisfy equation (5), the continuity equation. Another is that the internal flow is completely bounded. This means that the flow velocity is the same over the entire length of pipe,

$$\frac{\partial u}{\partial x} + \frac{\partial v}{\partial y} + \frac{\partial w}{\partial z} = 0 \quad \text{Eq. (5)}$$

where:

u, v, w , are velocities in the x, y, z directions, respectively.

In the case
pipe flow, v
and w are assumed
equal zero.

Figure (10)
a diagram of
the model of the
old water pipe
system.

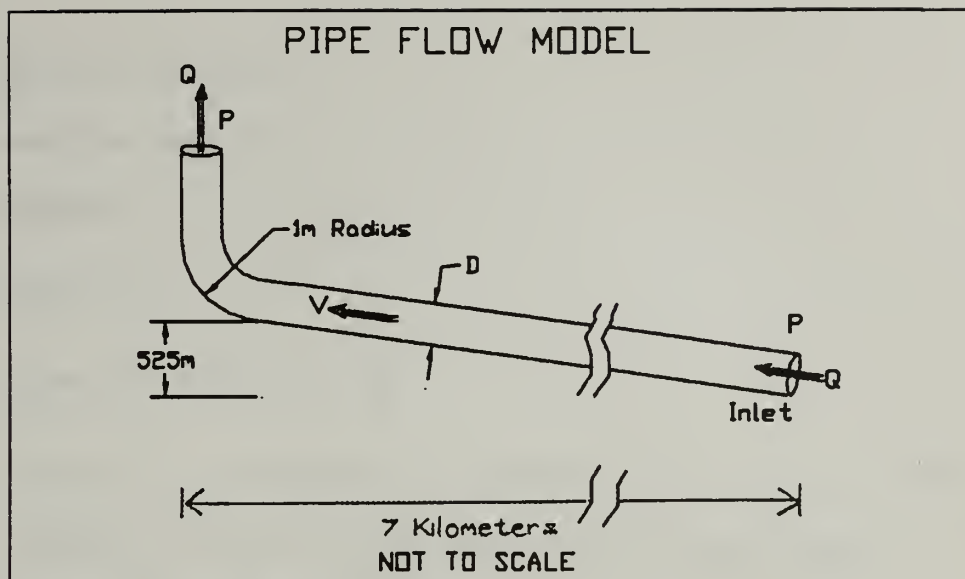


Fig. 10 - Pipe Flow Model

Head loss
over a length of the pipe can be expressed in the following equation:

$$\frac{P_1 - P_2}{\rho} = g (Z_2 - Z_1) + h_{l_T} \quad \text{Eq. (6)}$$

where:

h_{l_T} = Total head loss

P_1 = Initial pressure

P_2 = end pressure

Z_1 = Initial elevation

Z_2 = Final elevation

Head losses must be calculated differently depending on whether
the flow is laminar or turbulent. Pipe flow may be either laminar
or turbulent depending on the Reynolds number,

$$Re = \frac{\rho V D}{\mu} \quad \text{or} \quad \frac{V D}{\nu}$$

where:

ρ = density of fluid

\bar{V} = average velocity

D = diameter

μ = viscosity

ν = dynamic viscosity = $\frac{\mu}{\rho}$

$Re \leq 2300$ then the flow is assumed to be laminar. $Re > 2300$ is assumed to be turbulent. However, laminar flows may exist at much high Reynolds numbers under some conditions.

The change in pressure for laminar flow is computed analytically with the following equation:

$$\Delta P = \frac{128\mu L Q}{\pi D^4} = \frac{32L}{D} \cdot \frac{\mu \bar{V}}{D} \quad \text{Eq. (7)}$$

The head loss can be computed by:

$$h_l = \frac{32L}{D} \frac{\mu \bar{V}}{\rho D} - \frac{g(Z_2 - Z_1)}{\rho} \quad \text{Eq. (8)}$$

It can be seen after some algebra that:

$$\frac{32L}{D} \frac{\mu \bar{V}}{\rho D} = \frac{64}{Re} \frac{L}{D} \frac{V^2}{2} \quad \text{Eq. (9)}$$

where for laminar flow $\frac{64}{Re}$ is known as the friction factor.

As shown, for laminar flow, the pressure loss in the pipe is dependent on Reynolds number only, not on the roughness of the pipe surface.

For fully developed turbulent flow, the friction factor is determined experimentally. The pressure drop in this case is known to depend on the Reynolds number and the pipe roughness:

$$h_f = f \frac{L}{D} \frac{\bar{V}^2}{2} \quad \text{Eq. (10)}$$

where f is the friction factor

The relative roughness, e/D , must be determined from the pipe characteristics (see Appendix B-1). The friction factor can then be determined from the experimentally determined curves in Appendix B-2.

Head loss represents energy converted by frictional effects from mechanical to thermal energy. Head losses result from what are termed major and minor losses. Major losses include losses due to friction over the length of the pipe. This is the main concern with a pipe 7.2 m long. Minor losses are losses due to valves, tees, elbows and entrances. These are not considered here since the tees and elbows have not yet been determined.

Since the pipe diameter (inside) will be shown to be 1.086 m, flow velocity is expected to be 1.329 m/s, dynamic viscosity $1.56 \times 10^{-4} \text{ m}^2/\text{s}$, and $Re = 9.252 \times 10^5$. Therefore, the flow inside the pipe can be considered turbulent, and the losses are determined by Eq. (10). Since it has been established that turbulent flow is the case for the cold water pipe, Appendices B-1 and B-2 should be used. For HDPE pipe a value for e can be expected to be similar to that of commercial steel pipe. Appendix B-1 indicates that for such a pipe,

= .00015 and thus, for a 42 inch pipe $\frac{e}{D} < .0004$. Appendix B-2

en shows that for a Reynolds number of 9.252×10^5 , $f = .0167$. Head
ss due to friction in the pipe can then be calculated by Equation
0). Appendix A-1 shows the actual calculations.

PIPE WALL THICKNESS

The thickness of the pipe is crucial for lateral strength and
iffness since it must resist the wave and current forces acting
on it. The pipe must also resist pressure forces during the control-
d submergence during deployment and from the negative pressure from
e seawater pump(s), which could collapse the pipe if it is not
rong enough. The temperature gain in the cold seawater flow is also
function of the pipe thickness, since the only insulation of the
pe is expected to be from the pipe material itself.

With the pipe material determined, the required wall thickness
a be calculated from the forces acting on the pipe due to water mo-
on, and from the pressure forces acting on the pipe wall. The pipe
ckness must be great enough to prevent hydrostatic collapse. To de-
mine the critical buckling pressure for the pipe, the following
mula can be employed:

$$P = \frac{3EI}{(1-\mu^2) R_m^3} \cdot C \quad \text{Eq. (11)}$$

where:

P = Design pressure

E = Modulus of elasticity

μ = Poissons ratio

I = Moment of inertia

R_m = Mean radius

C = Reduction factor for out of roundness

(1.0 in this case)

In this calculation, the modulus of elasticity must factor in creep modulus, because plastic material creep under long term constant load. A common value to use here is $E_{50} = 200/.300 \text{ N/mm}^2$ (Brook 1981).

The moment of inertia in this case is calculated by the formula:

$$I = \frac{t^3}{12} \quad \text{Eq. (12)}$$

where t = pipe wall thickness

The calculation for the maximum stress and the maximum allowable stress is shown in Appendix A-2.

HDPE pipe is readily available in diameters up to 63 inches in foot sections. A pipe size of 1200 mm is selected for the seawater supply pipe. The manufacture's literature indicates that this size pipe is available in two wall thicknesses, indicated by SDR on the data sheet. SDR is the ratio of outside diameter to wall thickness. As shown by Driscopipe, a division of Phillips Petroleum, are 26 and 32.5; however, according to the sales representatives, other sizes are also available upon request.

The pipes used at Keahole Point vary in thickness. SDR's of 21 and 17 were used in areas where pressures were critical, or where resistance to abrasion on the sea floor was required. According to one source representative, the pipe can be manufactured at any thickness desired.

For the purpose of this design, an SDR of 21 is selected for the entire pipe length. The calculations in Appendix A-2 use an SDR of 21 and width of 1200 mm pipe size, which results in a wall thickness of 7 mm. The negative pressure required of the seawater pumps to overcome the major head loss due to friction is calculated to be $1.06 \times 10^5 \text{ N/m}^2$ (14.6 psi). This does not exceed the critical collapse pressure of the pipe, which is calculated to be $1.83 \times 10^5 \text{ N/m}^2$ (26.5 psi). Therefore, the thickness ratio is acceptable. Two other thicknesses were also examined. The results of these tests are as follows in Table (3).

Table (3) Comparison of Pipe Sizes and Thicknesses

Pipe OD m (in)	SDR (OD/t)	Wall Thickness m	δP_{crit} N/m^2 (psi)	ΔP N/m^2 (psi)
1.200 (47.2)	21	.057	1.83×10^5 (26.5)	1.006×10^5 (14.6)
1.067 (42.0)	21	.051	1.83×10^5 (26.5)	1.809×10^5 (26.2)
1.200 (47.2)	26	.046	9.39×10^4 (13.6)	9.098×10^4 (13.2)

The second and third pipe thicknesses examined had very narrow margins of safety. Although, these may be adequate, since the modulus

based on a 50 year life expectancy already, other stresses from bending and twisting of the pipe have yet to be considered. Therefore, the more conservative pipe design of 1.200 m OD and SDR 21 is selected.

The bending stresses on the pipe due to currents and wave motion could also be considered. Experience on other pipeline projects shows that these forces can be expected to deflect the pipe between the anchored sections and also during deployment. The maximum allowable deflection is given by the equations below.

$$\delta_{\max} = \frac{5w_0L^4}{384EI} \quad \text{inches} \quad \text{Eq. (13)}$$

I is given by:

$$I = \frac{M}{Z}(R_o^2 + R_i^2) \quad \text{Eq. (14)}$$

where:

I = Moment of inertia (in.⁴)

w₀ = Force per unit length (kips/inch)

E = Modulus of elasticity (kpsi)

R_o = Outside radius of pipe (inches)

R_i = Inside radius of pipe (inches)

M = Mass of the pipe section, $M = \rho \pi (R_o^2 - R_i^2)$ (lbs)

The actual calculation is left to a more detailed analysis and design.

TEMPERATURE GAINS

Calculating the temperature gain in the seawater flow could be as simple as determining the overall heat transfer coefficient for the pipe in seawater and using it in equation (15) to determine the heat transfer, then equating it with equation (16) to solve for the change in temperature,

$$q = U_o A_o [T_o - T_1] \quad \text{Eq. (15)}$$

$$q = M C_p \Delta T \quad \text{Eq. (16)}$$

However, the temperature of the seawater outside the pipe is not constant over the length of the pipe. This complicates the calculation, but experience from previous studies helps a great deal in the area of temperature gain.

In the Miami Beach study (Hirshman and Kirklin 1977), the extended cooling water temperature change calculation was based on a minimum flow rate of 50,000 gpm (3153 lps), a significantly higher flow than this Waikiki design. However, similar temperature and depth conditions to Waikiki make a comparison of methods reasonable. The water intake system was divided into several sections. External ambient thermal conditions for each section were determined from available data. Calculations for each section were completed using the same methods. Temperature gains in each segment of the system were added cumulatively to determine the delivered temperature. Calculations were repeated for different flow rates, and a minimum acceptable flow was determined.

The calculation was then initiated at the 525 m depth level, using an inlet water temperature of 6°C as the fluid temperature outside the pipe for the first segment (T_0) and the first segment length (L). This segmental thermal conduction temperature change was determined and added to the original inlet temperature. This new inlet temperature was used in the succeeding pipe segment temperature change calculation. Each succeeding incremental segment temperature change was obtained in this manner. The total temperature change due to conduction in the unburied section was then determined.

Almost the same method is used in the Waikiki case. To calculate the change in temperature the pipe length was divided into several sections. These sections were chosen from the bathymetry and temperature information and assigned average seawater temperatures outside the pipe. The temperature increase was calculated over the length of the first section of pipe and was added to the initial temperature of the next section of pipe for the calculation of that section, and so on. Six sections were chosen. A sample calculation is shown in Appendix A-3. No heat gain due to friction was considered in this case. Due to the experiences in other designs, it has not been a large factor.

Three assumptions were made to simplify this calculation. The first is that the force flow over the outside of the pipe from wave and current induced water motion is a constant 1 m/s (2 knots) in a direction perpendicular to the pipe in the heat transfer calculation.

ons. Actual measurements of steady current in the intended area of the project indicate a steady current of 1 knot is normal. A current of 2 knots is assumed to add a margin of safety to the calculations. For more refined calculations, a more accurate analysis of water motion over specific sections of pipe could be determined. This can be considered a "worst case" since we know also that the motion will probably not be perpendicular to the pipe everywhere. The second assumption is that C_p for seawater is constant over the length of the pipe, both inside and outside, even though from Appendix B-3 it is known to vary with temperature. A third assumption is that the pipe will be exposed to the surrounding seawater everywhere on its route. This will not be the case; some sections, such as in the surf zone and in the canal and harbor, will probably require burial. The assumption that convection due to flow of water around the entire circumference of the pipe is again a worst case assumption from a heat transfer standpoint for pipe laying on the bottom.

CALCULATION OF COLD WATER PUMPING POWER REQUIREMENT

The power requirement to pump the cold seawater can be calculated as follows:

$$\dot{W} = \rho \cdot h_1 \cdot V \cdot A \quad \text{Eq. (17)}$$

where:

\dot{W} = power required

ρ = seawater density

h_1 = head loss (note $h_1 \cdot \rho = \Delta P$)

V = flow velocity

A = cross sectional area of pipe (inside) = $\frac{\pi(ID^2)}{4}$

As calculated in both Appendices A-1 and A-2, the velocity is known to be 1.329 m/sec, ΔP is 6.022×10^4 N/m², and ID is 1.086 m, the power required is calculated to be 74,113 W. If a pump efficiency $\eta = 80\%$ is assumed,

$$\frac{74,113W}{\eta} = 92.64 \text{ kW}$$

$$1 \text{ hp} = 746 \text{ kW}$$

Therefore the pump must be at least 92.64 kW or 124.2 hp to deliver the seawater to the site at sea level.

The head loss and pressure drop considered here are only for the pipe losses. For each meter above sea level the water must be delivered, a simple calculation yields an additional 1,581 watts or 2.12 hp that will be required.

EFFECTS OF WAVES AND CURRENTS

The principal natural forces affecting a marine pipeline after installation are direct wave and current forces near the bottom. These can have secondary effects when, for example, the sediment under a pipeline is eroded and the pipeline becomes non-uniformly supported. The pipeline is considered stable if the forces holding it in place are greater than the forces that would tend to move the pipe. For an unanchored, unburied pipeline resting on a level, stable

bottom, the forces holding it in place are only gravity and friction.

Wave forces at the bottom are due to both a steady current and oscillatory motion produced by the waves. This motion reverses itself at half the wave period perpendicular to the wave front. In addition to regularly reversing drag and lift forces due to the horizontal water particle velocity, there is a horizontal inertial force due to the water particle periodic accelerations. The inertial forces are opposite in phase to the drag and lift forces. Each of these forces may be calculated separately, and whichever represents the larger force, since they are out of phase, at a particular depth is used as the maximum design force.

The forces experienced by a submarine pipe will vary depending on the depth of the water, currents, wave conditions, pipe size, shape, proximity to the bottom and to the free surface. The deeper sections of the pipe will see forces resulting from steady currents. The shallower sections will see forces resulting from wave action, as well as currents. Because of the differing conditions over the length of the pipe, several different systems of anchoring or supporting the pipeline may need to be employed to ensure stability (Brashman and Kirklin 1977).

The stability and structural integrity of the submerged pipeline depends, to a great extent, on its stability after installation. In the pipeline industry, the term stability refers to the ability of

e pipeline to remain as placed. If the pipeline moves or its surroundings are altered, its structural integrity is threatened. Much effort is expended in the pipeline design to predict these forces and then to design the pipeline for stability under the anticipated conditions.

The degree of stability required is a function of the pipe material used and its properties, with regard to resisting deformation or failure in tension, buckling, external or internal pressure. Plastic pipelines can withstand an amount of movement and deformation that would destroy a steel pipeline; however, the weight in water of the steel pipeline provides an inherent stability that is not available with the lighter plastic pipeline. This is a decision the designer must make early in the decision process. The decision has been made in the seawater air conditioning case to use HDPE.

It has been demonstrated by previous research (Hirshman and Kirklin 1977) that the forces caused by the design wave (hurricane conditions) greatly exceed the maximum effect of the design current out to depths of approximately 200 feet.

Considering winds, waves and currents, hurricane storm conditions represent the most serious long-term hazard to the submarine pipeline. A typical maximum deep water non-breaking wave of 50-foot height and 9 or 10-second period is commonly used as the design wave (Hirshman and Kirklin 1977). A maximum estimated bottom current of 2 m/s (1 m/s) is commonly used in conjunction with the wave forces,

less site specific data indicates otherwise. As stated earlier, a steady current of 2 knots is probably much higher than would actually be experienced by the pipeline, but such an assumption adds a large margin of safety to the design. Under hurricane conditions, this may actually be low.

There are some basic techniques used to increase the stability of the pipeline. These include increased weight of material, burial and/or other protective covering, and anchoring. It is common to bury the pipeline in shallow water where the forces and hazards, such as anchors, are maximum, and to lay the pipeline directly on the bottom in deeper water. Another technique, the catenary support, involves suspending a free floating, buoyant pipe with a flexible cable which is anchored to the bottom (Daniel 1989).

Reasonable assumptions for current and wave conditions for the purpose of calculations are a maximum bottom current of 2 knots running parallel to the coast, the design wave of 15 meters and a 10-second period in Appendix A-4. The calculations are made for a pipeline perpendicular to the coastline. The shelf and slope contours are assumed to be parallel to the coast, and the wave front in deep water is parallel to the pipeline.

In general, the forces acting on the pipe depend on both the pipe orientation, tides and wave conditions as stated previously. The basic hydrodynamics are derived from water particle kinematics. The equations for the particle motion in the horizontal plane are:

$$u_x = \frac{Hgk}{2\omega} \left[\frac{\cosh(k(d+z))}{\cosh(kd)} \right] \sin(\omega t) \quad \text{Eq. (18)}$$

$$\dot{u}_x = - \frac{Hgk}{2} \left[\frac{\cosh(k(d+z))}{\cosh(kd)} \right] \cos(\omega t) \quad \text{Eq. (19)}$$

where:

u_x = particle velocity in the horizontal direction

\dot{u}_x = particle acceleration in the horizontal direction

H = wave height

g = gravity

k = wave number

ω = angular velocity of wave, $\frac{2\pi}{\text{wavelength}}$

d = depth

These forces may act both in the horizontal and vertical directions. In addition to the hydrodynamic forces introduced by the motion of the water flowing around the outside of the pipe, the buoyant force of the pipe itself must be taken into account. Since the pipe's density is 0.954 g/cc (Rubin 1990), the buoyant force is a factor.

Description of Forces

Conventional engineering practice is to use the following equations to calculate the hydrodynamic forces on a pipeline associated with wave and current action (Verley, Lambrakos and Reed 1989). For a long cylinder, or pipe of diameter D submerged in a fluid of density ρ , the horizontal force, F_x , and the vertical force, F_z , are given

$$F_x = C_D \frac{\rho D l}{2} u |u| + C_M \rho \frac{\pi D^2}{4} l A \quad \text{Eq. (20)}$$

where

C_M = inertia coefficient

C_D = drag coefficient

u = undisturbed fluid velocity

A = fluid acceleration, \dot{u}_x or $\frac{du_x}{dt}$

$$F_z = C_L \frac{\rho}{2} D l u^2 \quad \text{Eq. (21)}$$

where

C_L = lift coefficient

Often the coefficients are assumed to be constant, but they are actually time dependent.

The first equation (Eq. 20) is known as Morrison's equation. The hydrodynamic coefficients can be taken from various references as determined from both potential theory and experimental data. The drag and lift equations utilize coefficients of drag, lift and inertia that are a function of pipe shape, Reynolds number, proximity to bottom and other factors. These are experimentally derived. The coefficients that are used are from a cylindrical pipeline resting directly on the bottom. Common values used in this situation are:

coefficient of drag, $C_D = 0.5$

coefficient of lift, $C_L = 1.4$

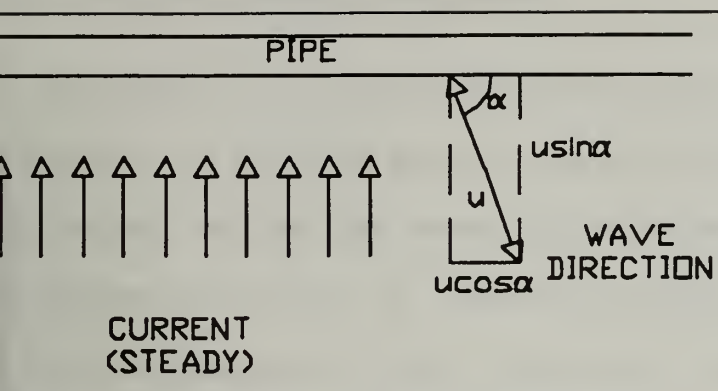
coefficient of inertia, $C_M = 3.3$

Experiments and field data have shown that these two equations predict the maximum hydrodynamic forces fairly well for the case of regular waves and a steady current. However, for the case of irregular waves and current, the variation between predicted and measured forces has been much greater. To describe the force time series for such combined flows, other models have been developed. For the purpose of this conceptual design, however, the basic model will be used.

It is important to understand how the different forces acting on the submerged pipeline interact. There is drag force in the lateral direction, consisting of both frictional and inertial effects of waves, and friction from currents. These are represented by the first and second terms in Eq. (20), respectively. There can also be a lift force on the pipe, as indicated by Eq. (21).

The orientation of the pipeline to the currents and direction of waves determines, to a large degree, what effect those forces have. If the pipe is oriented perpendicular to the flow of water over around it, those forces are greater. If it is oriented in a parallel direction to the currents and wave direction, the forces are greatly reduced. See Figure (11).

There are several parameters which may be considered in determining the hydrodynamic forces on the pipeline. Probably the most important one is the Keulegan-Carpenter number, $\frac{UT}{D}$, where U is the particle velocity, T is the wave period, and D is the pipe diameter.



g. 11 - Current and Wave Direction

If the Keulegan-Carpenter number is much less than 1, the inertial effects will dominate. This means that the water particle acceleration will be the major contributor to forces on the pipe. If the Keulegan-Carpen-

number is much greater than 1, the frictional or viscous effects will dominate, which means the water particle velocities will be the major contributor to the forces on the pipe.

There is a dependence of forces, due to passing waves, on the ratio of pipe diameter to period and wavelength of incident waves. Generally, in the case where $\frac{H}{D} < 1$ and $\frac{D}{L} \geq 0.15$, viscous forces can be ignored, and use of potential theory is recommended over equations (19) and (21). If $\frac{H}{D} \geq 1$ and $\frac{D}{L} \leq 0.15$, the empirically derived Morrison's equation (Eq. 20) can be applied. This relation considers the viscous effects. In cases where $\frac{H}{D} < 1$ and $\frac{D}{L} < 0.15$, other approaches can be used. With pipelines and other slender members, such as piles, in wave fields, the second case is relevant. This is why the industry standard is to apply Morrison's equation (Eq. 20) for the lateral forces.

Force Analysis of Various Configurations

In addition to the dimensions of the pipe with respect to incoming waves, the way the pipe is installed or configured can have a great effect on how the water flows around it, and hence, the resultant forces acting on it. There are many ways to install a pipeline, but for the purpose of this discussion, they can be broken down into five basic configurations as follows:

- (I) The pipe may lie flat on the bottom in some sections
- (II) The pipe may be partially buried
- (III) The pipe may be completely buried
- (IV) The pipe may lie on an uneven bottom with a small space under some sections
- (V) The pipe may be suspended above the sea floor, either on legs or free floating and anchored by cables. This is representative of the catenary support system.

It may be useful to look at each one of these configurations in shallow, intermediate and deep water. However, some of these can obviously be eliminated from analysis.

Determination of Forces for Configuration (I):

In the configuration shown in Figure (12), the pipe lies flat on the bottom. Although the design for the air conditioning system does not include this configuration, analysis of this type of installation demonstrates all the forces acting on each of the other basic configurations.

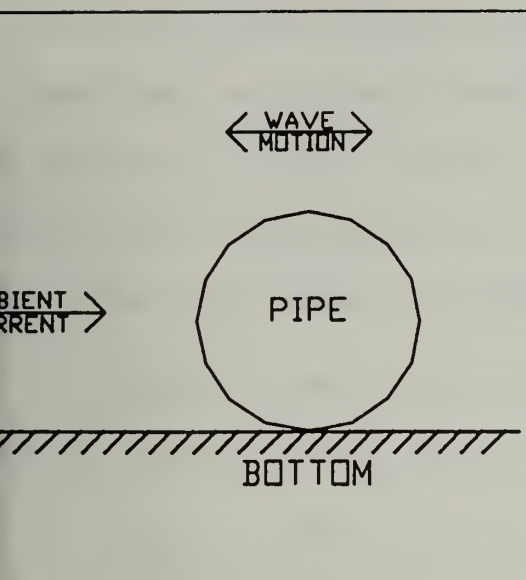


Fig. 12 - Configuration I,
Pipeline Flat on Bottom

ound the pipe can be broken down into two components: the component
duced by the wave action, and the
omponent induced by the steady cur-
nt. As previously stated, particle
ions due to wave motion in the hori-
tal direction are given by equa-
ns (18) and (19).

The motions are time dependent
90° out of phase as the wave
ses. It is important to see how the
agnitude of the viscous term in
(20), the drag portion, is depend-

on the square of the particle velocity given by Eq. (18). This is

rations. This is often considered the
worst case for designers. At this
point it is useful to look at a free
body diagram to illustrate how all the
forces act on the pipe. See Figure
(13).

Drag Force:

In this case, the drag force is
given by Morrison's equation (Eq. 20),
and as shown on the freebody diagram
(Fig. 13) as F_D . The flow over and

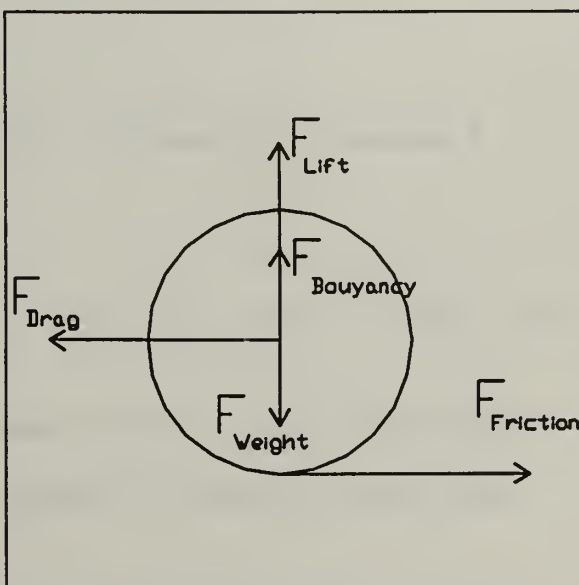


Fig. 13 - Free Body Diagram

contrast to the inertial term, the second term in Eq. (20), which dependent on the magnitude of the particle accelerations. Since the two terms in the lateral force are 90° out of phase, the maximum viscous force and drag force will not occur at the same time. In fact, they will occur at opposite quarter wave periods.

A steady current alone also creates a viscous drag force, but with no inertial term. This force is given by:

$$F_D = C_D \frac{\rho D l}{2} u |u| \quad \text{Eq. (22)}$$

As can be seen by the mathematical relationship between the drag force and the square of the velocities, the drag forces cannot be added directly to get the total force. Instead, the sum of the velocities should be substituted into Eq. (20). This equation then becomes:

$$\begin{aligned} F_x &= C_D \frac{\rho D l}{2} \cdot (u_{\text{wave}}(t) + u_{\text{current}}) \cdot |u_{\text{wave}}(t) + u_{\text{current}}| \\ &\quad + \text{inertial term} \\ &= C_D \frac{\rho D l}{2} \cdot (u_r(t)) \cdot |u_r(t)| + \text{inertial term} \quad \text{Eq. (23)} \end{aligned}$$

This term is part of the overall F_{drag} shown on the free body diagram (Fig. 13). The term l for pipe length is removed from the right side of the equation.

Lift Force:

The F_{lift} shown on Figure (13) also depends on both the wave generated particle velocities and the steady current. Again, as with the

ag forces, the wave incurred particle velocities are time dependent. Equation (21) is used, where $u = u_{\text{current}} + u_{\text{wave}}$.

Buoyant Force:

The buoyant force, F_{buoyancy} in Fig. (13), is calculated by multiplying the volume of the pipe by the density of the seawater it displaces. On a per unit length basis, only the cross sectional area is multiplied by the density as shown below:

$$F_B = \rho_{\text{water}} g \left(\frac{D_o}{2} \right)^2 \quad \text{Eq. (24)}$$

Weight Force:

The weight of the pipe, shown in Fig. (13) as F_{weight} , includes a component from the mass of the water in the pipe and the mass of the pipe material itself, as shown below:

$$F_{\text{weight}} = \rho_{\text{water}} g \pi \left(\frac{D_i}{2} \right)^2 + \rho_{\text{pipe}} g \pi \left(\frac{D_o^2 - D_i^2}{4} \right) \quad \text{Eq. (25)}$$

where:

D_i = inside diameter

D_o = outside diameter

ρ_{water} = seawater mass density

ρ_{pipe} = pipe material mass density

In the vertical direction, the forces consist of the buoyant force of the pipe and the opposing weight of the pipe, both of which are constant, and the lift force, which is dependent on time. The resultant of these is a normal force on the seafloor. If the normal force

negative in sign, however, the pipe would of course be accelerated toward the free surface. In that case, the normal force would have to be artificially increased in some manner for the pipe to remain stationary.

Bottom Friction Force:

The resultant normal vertical force is a key factor in the total horizontal force for the pipeline laying on the bottom, since the friction force between the bottom of the pipe and the surface of the seabed is dependent on the normal force. Therefore, this force is also time dependent.

$$F_{\text{friction}} = fF_N \quad \text{Eq. (26)}$$

where:

f = coefficient of friction

F_N = normal force

The coefficient of friction, f , differs for different bottom conditions and pipe intervals. This could be determined experimentally, but since bottom conditions can vary so much over the length of a pipeline, it is difficult to know what coefficient to use unless all conditions are known. Without specific data concerning the friction characteristics of the bottom, a conservative value for the coefficient of friction of 0.5 is used in industry for marine sediments (Brashman and Kirklin 1977). Thus, if the pipeline weighs 100 pounds per linear foot in water, it can be moved laterally on the bottom by a force of greater than 50 pounds per foot.

The sum of the forces in the horizontal direction includes the forces calculated from equation (20) plus the friction force calculated by equation (26).

To determine if the pipe is stable, all of the forces can be calculated at various time intervals over the wave cycle to determine the forces in the worst case, or maximum combined forces. If the pipe is determined not to be stable, concrete or other types of weight may be added to increase the downward force on the pipe, or some type of anchoring could be attempted to accomplish the same thing. Any weight added would result in increased weight minus the buoyancy of that material. Appendix A-4 includes a sample calculation of the forces acting on a pipe, configured as shown in Figure (12).

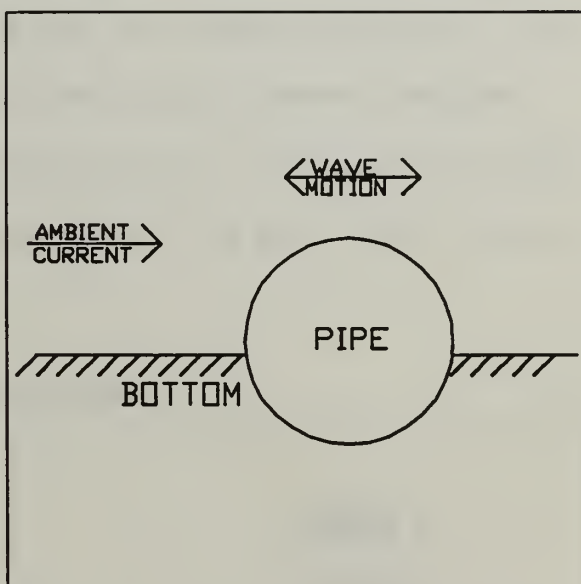
If the pipeline is anchored at regular intervals utilizing embedment-type anchors (into soil or rock), the computations are somewhat different than for stabilization solely by weight. In this case, the anchors prevent the pipeline from sliding horizontally. The spacing between the anchors for an unburied pipe is determined first by calculating the maximum vertical lift component of wave and current forces expected at a particular depth. The vertical force required for pull-out of each anchor set is determined, and a safety factor is applied. A maximum spacing calculation to prevent pull-out can also be determined. A further calculation is performed, using the maximum vector sum of wave and current forces, to determine, for a particular depth and anchor spacing, if the deflection of the pipe between anchors due to these forces can cause failure or damage to the pipe.

s is a function of the pipe material and design. The limiting safe distance between anchors can be either pull-out or deflection-limited, depending on circumstances at a particular depth.

Termination of Forces for Configuration (II):

A similar analysis of forces can be done on the pipe in Figure (II). The equations used in the first case can be used again with only slight modifications. The forces in the z-direction would change little. Lift force would be estimated in the same manner as would weight and buoyancy. Although the situation is not exactly the same, the coefficient of lift could be used to obtain a reasonably good estimate of the lift forces.

The inertia and drag forces in the x-directions, determined by equation (20), would be significantly reduced. The viscous drag term could be reduced by a factor of 2 since only $\frac{1}{2}$ of the diameter D is exposed to the particle motions. The reduction in diameter has an even greater effect on the inertial term since, in that case, the diameter is squared. It is immediately obvious that the frictional force may be greater, depending on how deeply the pipe may be buried.

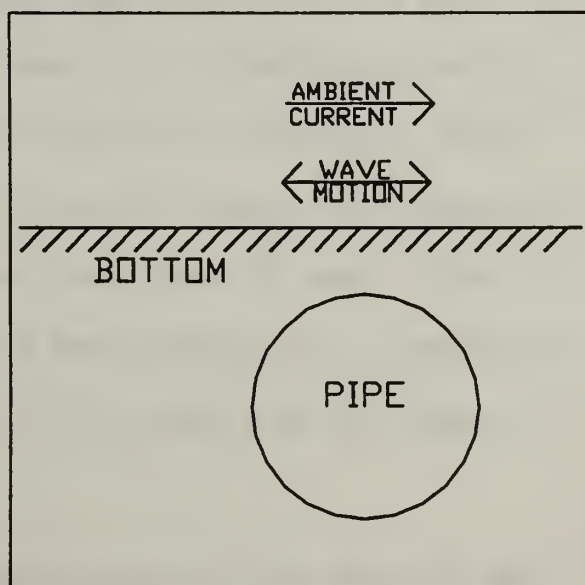


**Fig. 14 - Configuration II,
Pipeline Partially Buried**

Partial burial is not anticipated to occur in the design of the ventilation center air conditioning system; however it is a common configuration of submarine pipelines. This is often different than the original configurations of the pipeline, since the action of the waves and current may redistribute the seabed around the pipe. The forces required for break-out of the pipe in the semi-buried state depend on soil characteristics and permeability, as well as the characteristics of the hydrodynamic forces. The break-out of the pipeline relies on a couple of mechanisms. If the wave induced lift force and buoyancy force exceed the weight of the pipe, an upward force will result. At this point, there are two forces resisting upward response of the pipe; these are the shear stress between the pipe and the sand sediment grains, and the negative pressure of the pore water below the pipe (Foda, Chang and Law 1990).

Termination of Forces for Configuration (III):

In case III, the pipe is completely buried (see Fig. (15)). The design of the air conditioning cold water supply pipe is intended to include burial of the pipe in the canal, at harbor and through the surf zone. This is considered the best way to avoid damage to the pipeline. The hydrodynamic forces are less relevant when they are reduced to only the



**Fig. 15 - Configuration III,
Pipeline Completely Buried**

buoyancy and the weight of the pipe; it must simply be heavy enough to float in the sediment. The forces here also may vary depending on the qualities of the bottom. The bottom sediment could liquefy under some wave conditions. This case requires the study of seafloor mechanics.

Determination of Forces for Configuration (IV):

In case IV, the pipe is suspended on top of some sort of support structure (see Fig. 16). This is the intended method of pipe installation

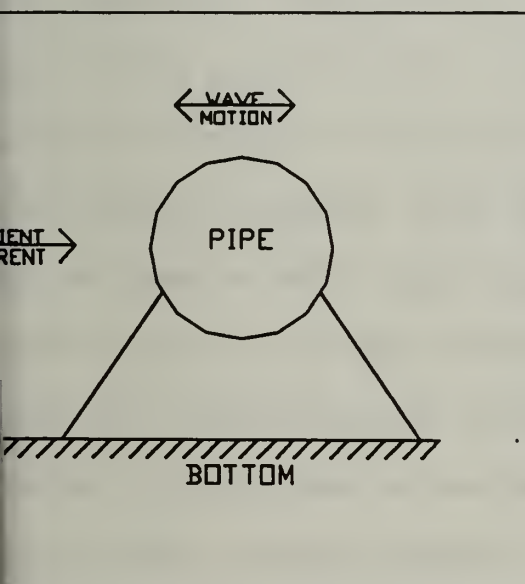


Fig. 16 - Configuration IV, Pipeline Supported Above Seafloor

off Waikiki beyond the surf zone out to a depth of approximately 60 meters. If the pipe can be supported one diameter or more above the sea floor, the lift force can be neglected. The reason for this is if the particle motion around the bottom of the pipe is nearly identical to the particle motion over the top of the pipe, then

the two forces are equal and opposite, with a net resultant of zero. This is neglecting any difference in particle velocities between the depth at the top of the pipe and the depth of the bottom.

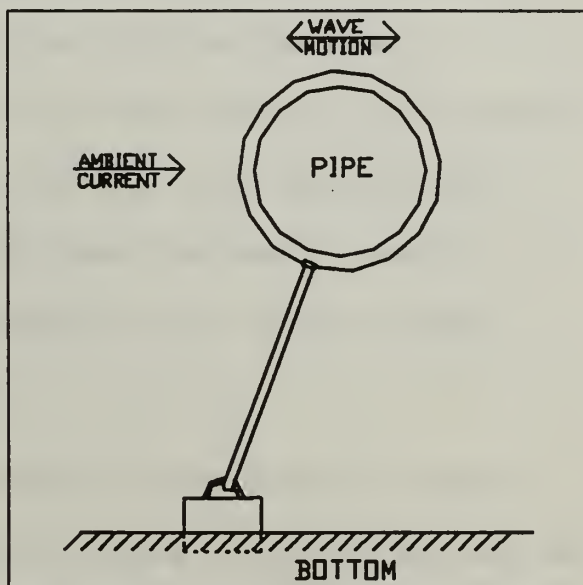
The supports would probably be concrete, as in the case at Kea Point. They serve dual purposes by adding weight and reducing

t. In this case the normal force resulting from the difference between combined buoyancy and the combined weight of the pipe and support is the total normal force used to determine the friction force, which acts to resist the drag and inertial forces in the lateral direction (x-direction).

Determination of Forces for Configuration (V):

In the case V, as in case IV, lift forces on the top are negated by the identical force acting on the bottom of the pipe (see Fig. 17). This type of support is representative of the catenary support used successfully at Keahole Point. This type of support has all the same advantages as case IV. One additional force to consider in this case is the forces acting on the flexible support. These must be calculated in a similar fashion to the forces on the pipe.

Again as in the fourth case (Fig. 16), the weight of the anchor acts to overcome the excess buoyancy force in the z-direction and the drag and any inertial forces in the x-direction. This type of support would probably only be used in deep water, therefore the forces would result from the steady current present. With no particle veloci-



*Fig. 17 - Configuration V,
Catenary Mooring of Pipeline*

s and accelerations from wave action, the steady velocity is only from the current, and the inertial term is neglected.

Picking coefficients such as the drag coefficient C_D and added mass coefficient for the forces in the x-direction can also be difficult. Just as in the case of lift, the forces affecting the pipeline are dependent on distance from the bottom, depth, marine growth, current conditions and wave conditions. Different values for the coefficients could be determined experimentally for all the various conditions, and researchers have tried to do this. Some of the experiments done in the area focus on finding the RMS (root mean square) values of those coefficients for given Keulegan-Carpenter numbers. In another approach, the forces can be decomposed into Fourier components (Shankor, Cheong and Subbiah 1988).

There is a great deal of data published on the subject, with a number of different approaches to determine these coefficients and improving the predictions for Morrison's equation and variations of the equation. Many contradict the predictions of the Morrison's equation, and most of these are not as easy to use.

Scour

A pipeline laying on the seabed may lie flat or there could easily be areas under the pipeline where flow could pass due to the uneven seabed. These areas could be subject to erosion of the seabed due to the water motion and scour. Scour under and around the pipeline can alter the seabed upon which the pipeline relies for stabil-

y. Prediction of scour below and around pipelines is a major focus of pipeline engineering practice. This is an important area to consider when a pipeline is to be exposed to currents or waves, and potential theory is unable to explain or predict scour depth (Sumer, Hansen and Fredsøe 1988).

In a steady flow, the scour can be expected under and on the leeward side of the pipe, but in oscillatory flow the scour of one half period is the upstream part of the next half period. The scour depth depends on several factors. The size and specific gravity of the sediment particles which comprise the seabed play a role, as does the consolidation of the bed, just as in the case of longshore transport.

Although water particle motions from a steady current and wave motion passing beneath the pipe may exceed the critical velocity required for the onset of sediment transport in the vicinity of the pipe, this is not required for scour to occur. The focus of the studies of scour around pipelines is the vortex shedding on the leeward side of the pipe (see Fig. 18).

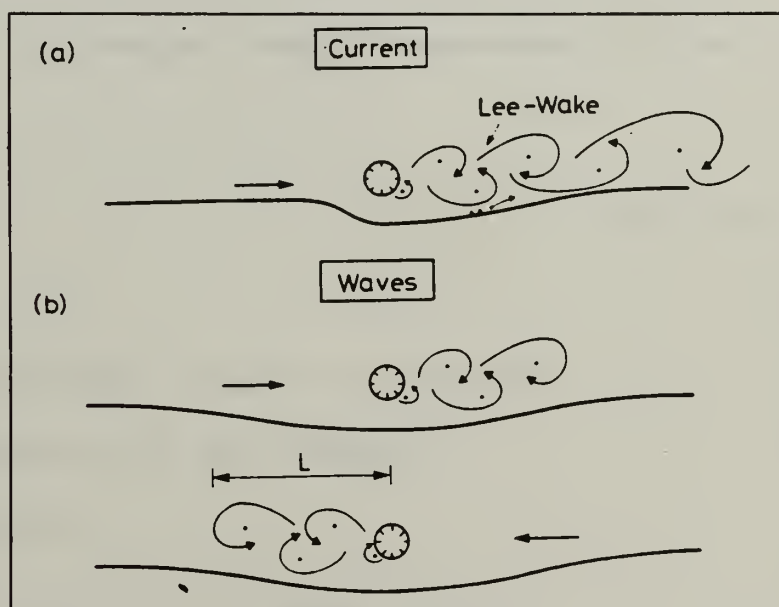


Fig. 18 - Lee-Wake Effect:
(a) Currents, (b) Waves

As the flow exceeds the critical Reynolds number and flow around the pipe becomes turbulent, vortices form on the leeward side of the pipe. As the flow velocity increases, the vortices begin to shed into the wake in an oscillatory manner. The velocity of the particles in the vortices can then exceed the critical velocity required for the onset of sediment transport.

The vortices can be characterized by St , the Strouhal number:

$$St = fD/U \quad \text{Eq. (27)}$$

where:

U = the mean flow velocity

f = the vortex shedding frequency

D = pipe diameter

The shields parameter θ_o is another useful measurement of the flow around the pipe:

$$\theta_o = \frac{U_f^2}{g (s-1) d} \quad \text{Eq. (28)}$$

where:

U_f = undisturbed shear velocity on the bed

s = specific gravity of the sediment

d = sediment size

g = gravity

The depth and position of the scour depend on the Strouhal number, shield parameter, height of the pipe from the seafloor and soil characteristics.

The Keulegan-Carpenter number is considered the governing parameter of the scour trench depth (Sumer and Fredsøe 1990 and Chiew 1993). Waves and currents affect scour differently as Figure 9) illustrates.

Sumer and Fredsøe (1990) present an relation for a pipe control with a bed,

$$\frac{S}{D} = 0.1\sqrt{KC} \quad \text{Eq. (29)}$$

where:

S = final scour depth below

the pipe (max)

D = diameter of pipe

KC = Keulegan-Carpenter

$$\text{number} = \frac{UT}{D}$$

The trench reaches an equilibrium depth when it is large enough that the flow velocities are decreased to below the critical velocities for initial scour (θ_{crit}). As the Keulegan-Carpenter number increases, the scour increases both in depth and length. Roughness of pipe has almost no effect. The scour beneath the pipe is known as

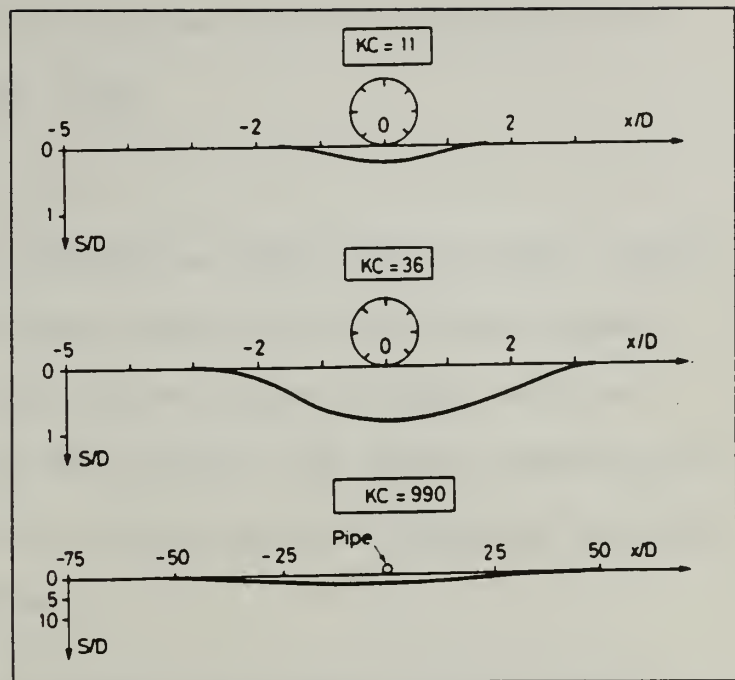


Fig. 19 - Equilibrium Scour Profiles for $KC = 11$, $KC = 36$ and $KC = 990$

the tunnel scour. The scour occurring downstream of the pipeline is known as the lee-wake scour (Chiew 1993).

Vibrations

Not only can the bed around a pipeline erode due to scour, vibrations of the pipe are induced by both wave action and vortex shedding. The wave action produces lift forces which, as seen earlier, oscillate in amplitude during the wave period. The vortex shedding induces changes in pressure acting on the pipe with a frequency at that of the oscillations of the vortices.

There may also be significant changes in the force coefficients, C_D and C_M due to the vibrations induced on a pipe. The drag coefficients tend to be larger for vibrating cylinders, partly due to an apparent increase in diameter and partly due to an increase in the velocity. The inertia coefficient, C_M , decreases relative to a fixed pipe (not vibrating) attributable to the pipe hitting its own wake. The effect of the coefficient of lift does not appear to be significant (Sumer, Fredsøe, Gravesen and Bruschi 1989).

Discussion of Combined Effects/Pipe Design

The industry standard for predicting forces in submarine pipelines is the use of Morrison's equation (Eq. (20)) and Eq. (21) for lift forces. However, there are serious doubts as to the accuracy of this approach when used alone. The use of equations (20) and (21) is also made more difficult in some pipe configurations, such as par-

ally buried pipelines; they are not applicable at all to buried pipelines.

The effect of scour and vibration should also be taken into account when designing a pipeline. Not only should the forces and responses for intended installation configurations be considered, but the forces and responses associated with any anticipated changes in the configuration should be taken into account as well. Tunnel scour and lee-wake scour contribute to changes in the configuration of the pipe and vibration due to vortex shedding could add to the effect.

Empirical and analytical analysis of a design should be supported with model testing, if possible, before going forward with a design. Taking all the force considerations into account, the pipeline design should include three of the configurations mentioned.

The nearshore sections of the cold water supply pipe and effluent pipe located inside the reef, in the Ala Wai Boat Harbor and Canal, will all be buried. This would require a significant dredging effort. Since the channel is already dredged for navigation, the pipe in the boat channel would have to be buried to a depth at which future maintenance dredging for the the channel would not damage the pipe. Along with burial, some form of securing the pipe to the bottom of the trench is advised. This could be accomplished by bolting the pipe down with clamps if there is sufficient basaltic rock or coral growth into which bolts could be driven. Another method could be to weight the pipe down with concrete. This section of pipeline would be

a total length of 2500 meters, extending out to a depth of approximately 20 meters.

From the 20 meter depth to approximately a 100 meter depth, the pipeline could be anchored by concrete blocks attached to the pipe. This method was used successfully at this depth at Keahole Point. The blocks and anchors would support the pipe at least one diameter above the sea floor to minimize the hydrodynamic lift. A calculation similar to that in Appendix A-4 could determine the amount of concrete per unit length of pipe which would be required for the pipe to remain stable for this section. This section would extend approximately 900 to 950 meters offshore, beyond the buried section.

The remaining section of cold water supply pipe could use the unique buoyant catenary support, as designed by Makai Ocean Engineering Inc., Waimanalo, Hawaii to avoid the rough bathymetry of the steeper sections. This was used at depths of 160 meters and greater at Keahole Point, and has proven successful. Off Waikiki however, the sharpest drop off begins at about 100 meters. Considering bottom contour, steepness, and roughness of the bathymetry, this is the desired depth at which the use of catenary supports should be initiated.

This could be a hazardous area in which to make a transition to the more fragile catenary support, depending on the wave conditions encountered at this section. At this distance offshore, waves may encounter the pipe section at adverse angles (close to perpendicular). Calculating the wave forces on the pipeline by the

method in appendix A-4 however, indicates that the lateral force acting on the pipe due to the incident design wave is negligible. This is to be expected since the design wave of height 15 meters and period of 9 seconds would have a wavelength of 126 meters and in 100 meters of water, these are deep water waves.

The details of the bathymetry are important to the decision where to make the change to the catenary support. A small change in the route of the pipeline could change the depth at which the catenary can be safely employed. For these reasons, a particularly thorough investigation of the bathymetry and bottom conditions is recommended along this section of the pipe route prior to final design.

ENVIRONMENTAL CONSIDERATIONS

The seawater cooling concept calls for a single pass of the cold water through the seawater-chilled water heat exchanger, raising the temperature of the seawater some 5⁰C. Once through the heat exchanger, the seawater is essentially high quality waste water, still cooler than ambient surface water. In the design of the system, non-toxic and mostly non-metallic materials are utilized. The discharge water is clean seawater, with excellent water quality characteristics.

Discharge outfalls from the convention center to the ocean would produce no net positive or negative environmental impact because of small discharge volume compared to the size of the receiver body and the good mixing in the nearshore region. Using a diffuser at the discharge end of the outfall would bring the water to within a few degrees of ambient at the discharge point, and since the quality of the discharge water is excellent, no adverse effect is anticipated. The outfalls, however, will be expensive because they will have to be designed to withstand the rigors of the surf zone and will fall into the expensive offshore construction category.

FOULING

In the ocean intake pipeline, fouling in the offshore pipeline is not expected to significantly reduce flow efficiency. The cold water is nutrient rich, however, research at Keahole Point, Hawaii, has shown that bio-fouling will only occur if the water is exposed to sunlight. Exposure of the water to sunlight is avoided in this design. Severe internal fouling is not likely to occur unless the system is shut down for a long time, in which case fresh water or dechlorinated water could be sealed in the pipeline. Often, just sealing the pipe causes the system to turn anaerobic, killing the aerobic organisms. The pipeline could permit cleaning "pigs" of the type used in oil pipelines or other devices, to be pulled through. However, this is not expected to ever be necessary.

ALTERNATE DESIGN: DIRECTIONAL DRILLING

An alternate technology which could be employed in lieu of dredging the nearshore sections of pipeline is directional drilling. This employs machine drilling equipment to drill shafts which may run horizontally or at any desired angle for as much as a mile. The technology currently allows the drilling machine to be guided in a manner such that it can curve upward and outward and break through the surface at any desired target point.

Directional drilling has been in use in the gas and oil pipeline industry for twenty years, and more recently used by the telecommunications industry.

Directional drilling may be a desirable method whenever the environmental or economic impacts of open drilling are unacceptable. Shafts can be drilled under highways, industrial areas, rivers or any surface obstacle which cannot be disturbed.

The equipment used is similar to standard drilling equipment and employs sensors in the drill bit to measure compass heading and angles of tilt and rotation. The computer guided drill head is accurate within 1% of the shaft length (Wilkins, Muller 1991).

The problem which first comes to mind is the inhomogeneities which exist in the volcanic terrain, which makes up the Hawaiian geology. However, this and many other technical problems were overcome in demonstration of directional drilling at Keahole Point, Hawaii

February, 1992. At that time, a shaft of 3.75 inch diameter was successfully drilled a distance of 610 feet through inhomogeneous volcanic rock to a point 320 feet offshore, at a depth of 30 feet, where it broke out. The test shaft was then reamed out to a diameter of 26 inches, for a distance of 20 feet when the demonstration was considered a success. It is believed that shafts could be reamed out to a diameter of as much as 54 inches with the present technology.

The drilling technology could conceivably be used in the near-shore surf zone for both the cold water supply pipe and the effluent pipe. The shafts could still be lined with the 1200 mm diameter high density polyethylene pipe. This would necessitate some type of joint between the offshore section of the pipe and the drilled section.

The advantages would be the superior pipe protection from current and wave forces on the pipeline in the surf zone. It might also alleviate many of the environmental problems associated with dredging a trench for the pipeline. Any dredging activity in that area would be subject to intense scrutiny from environmentalists and the visitor industry in Waikiki.

However, there are disadvantages as well. The drilled shaft would probably have to begin at the convention center site itself, due to space constraints in the area. This would necessitate a drilled shaft length of approximately 2500 meters or more. This is longer than has been proven technically feasible to date.

it may not be possible. Another disadvantage is the expense. The cost to drill a tunnel of that length and diameter would be much greater than the more conventional alternative.

Although the drilling technology is an interesting option, at this point, there are too many unknowns to propose it as a viable method of pipeline installation in this application. For these reasons, the more conventional methods of installing the pipeline are intended for use on this project.

System Design

SEAWATER HEAT EXCHANGERS

Heat exchangers are available in a variety of types, sizes and materials. There are standard models available by many manufacturers, however, a single unit of the size required for an air conditioning application of this size would probably be custom made.

One available design is called a finned tube heat exchanger. The seawater could pass through tubes that have fins attached over which the fresh water would pass. The tubes or coils could be constructed of titanium, thus resisting corrosion by the seawater, and the fins could be aluminum.

Another design considered is a plate and frame heat exchanger. These are made of an assembly of pressed metal plates, aligned on or secured to metal frames or bars, and a containment or cover. Gaskets are set in the outside grooves to contain the fluids and to direct the fluid flow distribution. The hot and cold fluids can flow in the grooves in the plate in opposite sides of the plates. This is the design selected in this air conditioning design.

There are many advantages to plate and frame heat exchangers. The principle advantage is the accessibility of the heat exchange surfaces for cleaning. Other advantages claimed by plate and frame heat exchangers over tube and shell exchangers are as follows:

1. Less surface area required for heat transfer, resulting in cost savings. The close spacing of the

grooves between adjacent plates result in a geometry similar to that of very small tubes.

2. Less space required for the unit(s) as a result of the reduced surface area.
3. Better heat exchanger effectiveness.
4. The number of plates can be easily increased or decreased depending on future load changes.

The only disadvantage over other types is the large number of faces sealed by gaskets. Due to the use of elastomeric materials in the gaskets, the internal temperature and pressures of the heat exchanger are generally limited to 300 °F and 400 psi; however, both are well within the range of this air conditioning application.

Materials

In a series of experiments and corrosion tests at NELH, ALCAN International has determined that the heat exchangers for deep cold water application can be constructed of aluminum alloy. The water pitting corrosion, which is normally the result when exposed to 6°C seawater, can be controlled by introducing a thin surface layer of zinc to the aluminum. This limits the pitting to the zinc layer. Heat exchangers constructed of this zinc clad aluminum alloy are less expensive than the titanium units and are expected to last 10 to 20 years.

Bio-fouling

One of the major concerns with seawater heat exchangers has been bio-fouling. However, as with the cold water supply pipe, research has shown that bio-fouling on the cold water side of the heat exchanger is virtually nonexistent as long as the cold, nutrient rich seawater is not exposed to sunlight.

Design

The type of heat exchanger, the material used, the heat transfer required, and the temperature differences of the fluids on both the "hot" and "cold" sides determine the size and cost of the unit or units.

The size is determined by the surface area required over which the fluids must flow. The amount of thermal energy transferred is determined by:

$$q = K \cdot A \cdot \Delta T_m \quad \text{Eq. (30)}$$

where:

q = the desired amount of thermal energy transferred

A = the heat transfer area

K = the overall heat transfer coefficient

ΔT_m = the log mean temperature difference (LMTD)

The desired thermal energy transferred is the result of the change in temperature desired in the fluids multiplied by their spe-

ic heats and the mass flow, \dot{m} ,

$$q = \dot{m} \cdot C_p \cdot (T_{in} - T_{out}) \quad \text{Eq. (31)}$$

The effective area of the heat transfer is a function of the geometry of the heat exchanger. It represents the surface area exposed to the fluid.

The overall heat transfer coefficient, K , is also a function of the geometry, and the materials and fluids used.

is a characteristic specific to the application, and is often determined by pre-

vious experience. This is adjusted down for a fouling factor which accounts for the reduction in effectiveness over time due to fouling of the grooves or tubes in the heat exchanger.

The log mean temperature difference is the temperature parameter which heat exchanger performance is based. It is defined by the temperature difference at the other end of the heat exchanger divided by the natural logarithm of the ratio between the two temperatures. The temperatures are illustrated in Figure (20) for a counterflow heat exchanger.

LMTD is calculated as follows:

$$LMTD = \frac{(T_{H2} - T_{C2}) - (T_{H1} - T_{C1})}{\ln \left[\frac{(T_{H2} - T_{C2})}{(T_{H1} - T_{C1})} \right]} \quad \text{Eq. (32)}$$

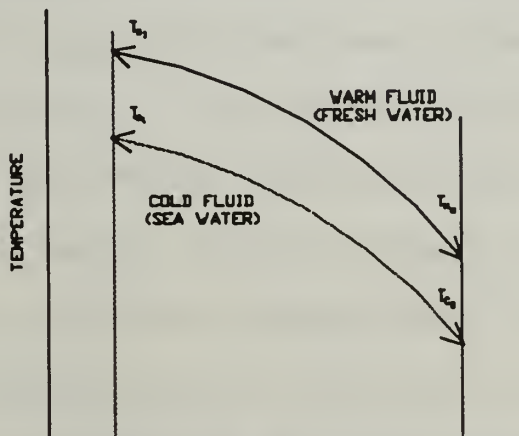


Fig. 20 - Log Mean Temperature Difference

In the case of the seawater air conditioning system, the LMTD is all, about 0.95 °C. Therefore, a high heat transfer coefficient is desirable to reduce the area required as much as possible.

Zinc-clad aluminum plate heat exchangers of the type required would probably have to be custom built and custom designed. Titanium heat exchangers are available "off the shelf" from companies such as Alfa-Laval Marine & Power and Mueller Accu-Therm. Aluminum alloy heat exchangers are also available "off the shelf." ALCAN of Canada is one manufacturer which has had success with aluminum seawater heat exchangers at Keahole Point, Hawaii. Aluminum has the advantage of being much less expensive than titanium, however, it does not last as long. Life expectancy for aluminum alloy heat exchangers might be about 20 years, as opposed to 30 or 40 years for titanium. It can be expected that aluminum heat exchangers and titanium heat exchangers would be in competition for a job of this size, and economics would decide which type and models would be chosen, provided they could adequately meet the performance requirements. Table (4) summarizes the heat exchanger requirements. Such data can be supplied to vendors and manufacturers who could respond with options for "off the shelf" designs in the case of titanium heat exchangers.

Table (4) Heat Exchanger Design Criteria

	Hot Side	Cold Side	Units
Media	Water	Seawater	
Volumetric Rate	19,511/1.231	19,511/1.231	GPM / m ³ /s
Mass Flow Rate	162,727/1230	167,215/1265	lb./min / kg/s
T _{in}	54.7/12.6	43.7/6.5	°F / °C
T _{out}	45.3/7.4	52.9/11.6	°F / °C
Density	8.34/999	8.57/1027.6	lb./gal / kg/m ³
LMTD 1.7°F / 0.95°C Load 86,677 x 10 ³ BTU/hr / 2.54 x 10 ⁴ kW			

The coefficient of heat transfer or operating K value for titanium heat exchanger can be expected to be about 7166.4 W/m²°C or 1262 Btu/ft².h.°F.

Using Equation (30), with the load of 2.52 x 10⁴ kW,

$$2.52 \times 10^4 \text{ kW} = 7166.4 \text{ W/m}^2\text{°C} \cdot 0.95\text{°C} \cdot \text{Area}$$

Yields 3702 m² for the effective surface area of the heat exchanger. This is larger than any previously known heat exchangers for this purpose, but based on information on other "off the shelf" models, a single unit of this capacity might occupy 66 cubic meters of space in a facility. It is unlikely that a single unit would be used. It might be wise to have several heat exchangers and excess capacity so that one could be shut down for cleaning and maintenance without loss of air conditioning capacity. On the other hand, since the entire air

conditioning system is intended to be backed up by a conventional system, minimum redundancy in the heat exchangers is required.

Even without redundancy, space, weight and transportation constraints might restrict the size. Therefore, it is assumed here that three units would be installed in parallel, each of effective area 14 m^2 and possibly containing as many as 1,000 plates. This is only an estimate of what would be required; the units vary from manufacturer to manufacturer.

PUMPS

The design of the pumping system includes an onshore sump for the pumps. The sump may be wet or dry. The onshore sump would provide access for maintenance. The construction of the sump could be adjacent to the Ala Wai Canal, and only a short trench or tunnel would be required.

The pumps themselves could be three or four centrifugal pumps, arranged in parallel, to provide the required discharge head of 58.63 meters. Variable speed pumps could be used, and controls could be set in a manner which allows only one or many pumps to be used on the seawater side at one time. The flow rate could be controlled automatically by a temperature sensor on the discharge side of the heat exchanger. This would reduce power consumption during off peak times.

The pressure drop in the heat exchanger is of great importance. The water area required by the heat exchangers increases the pressure

p which occurs and increases the pumping power required to force
h the seawater on the cold side and the fresh water on the air con-
ioning side through the heat exchanger at the desired flow rates.
e power required for this task can be expected to be several times
her than the power required to pump the cold water to the site.
y 93 kW out of a projected 720 kW is required for seawater deliv-
. Most of the remaining 617 kW power requirement is for the head
s in the heat exchangers. The heat exchangers could be expected to
sent a pressure loss of as much as 83 KPA or 12 psi.

Economic Analysis

GENERAL APPROACH

The intent of this design is that a conventional air conditioning system be installed as both a back-up system and for contingency loading in the event the complex requires supplementation during peak load periods, or due to unforeseen future facility expansion. Therefore, the capital cost of the proposed system is the same as a conventional system plus the added expense of the seawater loop. The total additional capital costs analyzed are only those associated with the construction of the large heat exchangers, the cold water supply and discharge pipe, the related pumps and control system as shown in Table (5). Some of the assumptions made, such as the inclusion of a conventional back-up air conditioning system, are extremely conservative. In subsequent design iterations, such overkill would probably be eliminated in the name of cost cutting.

ESTIMATE OF CAPITAL COSTS

Costs of pumps and controls are difficult to estimate without detailed mechanical drawings. The estimate of \$750 K for materials and 100 K for labor, equipment and installation is a rough guess based on estimates of other similar projects. The accepted method of estimating cost of the pipeline material is by weight. A cost of \$1.00 per pound of HDPE is used. This price has been fairly stable over the past several years, although recently, due to an "oil glut," the prices have dipped as low as \$.80/lb. According to the vendors, how-

er, this is not expected to last. The cost of \$431 per meter of
 e is arrived at by using the pipe's diameter of 1200 mm and thick-
 ss of 57 mm to calculate the volume of material in 1 meter of pipe,
 en multiplying that by density of 956 kg/m³ and the cost of \$1/lb.
 2.21 lbs./kg. The assumption made earlier was that the pipe would
 of uniform thickness throughout the cold water loop. The nearshore
 ction of 2600 meters length and offshore section of 4600 meters
 ngth are then multiplied by the cost per meter of \$431.

Table (5) Capital Costs of Seawater Air Conditioning System
(all costs are loaded, including overhead and profit)

Item Description		Material Cost			Construction Cost (Labor & Equipment) (thousands \$)	Total Cost (thousands \$)
		Unit Cost (thousands \$)	Quantity	Total Material Cost (thousands \$)		
Pumps and Controls			LS	750	500	1,250
Seawater Supply Pipe	Nearshore Sections	0.431/m	2,600	1,121	4,480	5,601
	Offshore Sections	0.431/m	4600	1,983	2,974.5	4,957.5
Effluent Pipe (all nearshore) redundant effort with supply pipe)		0.431/m	2,600	1,121	1,121	2,242
Heat Exchanger		450	3	1,350	250	1,600
& Misc. Plumbing		100	LS	100	included	100
Subtotals				6,425	9,325.5	15,750
Contingency 10%				643	933	1,570
Permits/EIS		?	?	?	?	?
Totals				7,067	10,258	17,325.5

The construction costs (labor and equipment) for the pipeline are more difficult to estimate. According to Makai Ocean Engineering, Waimanalo, Hawaii, their experience in designing the pipelines at Keahole Point has been that the overall cost of the pipelines are generally five times the cost of the material for the pipeline. Therefore, labor and equipment costs can be estimated to be about four times the material price (Van Ryzin, Leraand 1991). Other estimates of the Keahole Point project indicate that the cost of the offshore section may be a little lower since there is no excavation involved. The multiplication factor may therefore be reduced to 3.5 for the offshore section (Syed, Nihous and Vega 1991).

In this case the multiplication factor is lowered further for several reasons, most of which relate to economy of scale. Previous estimates were based on much shorter pipelines. The cost of setting up a staging and assembly area, building forms for pre-cast concrete anchors and bringing a large tugboat in from the mainland would all be examples of large one time set-up costs. Spreading these costs over the length of the offshore section of the pipeline could significantly reduce the cost. Therefore the total multiplication factor used is 2.5 in this estimate.

The figures in Table (5) for construction costs for the two sections of supply pipe are arrived at by using multiplication factors of 5 and 2.5 for the nearshore and offshore, respectively. The multiplication factor for the effluent pipe is reduced to 2, instead of 5,

cause of much duplication of effort in the nearshore zone for the
fluent and supply pipelines.

The cost of the heat exchanger is generally 25% of the cost of
the overall system (Van Ryzin 1991). Using that as a basis, the heat
exchanger and related installation might be expected to cost about
\$5 M. However, these estimates were based on systems with large on-
shore distribution systems and supply pipes less than one third the
length of the one in this design.

Based on 1991 costs quoted on smaller titanium heat exchangers,
the heat exchanger requirements could be met at a cost of approxi-
mately \$450,000 each, for three units including freight and taxes. An
additional \$100,000 is added to this for miscellaneous valves and
plumbing materials. Labor and equipment costs for installation and
testing could run another \$250,000. The total heat exchanger esti-
mate, including miscellaneous plumbing, is \$1.7 M. It should be noted
that the use of an aluminum alloy heat exchanger would reduce
the cost of this item considerably. Such a cost cutting option would
most certainly be considered in subsequent design iterations.

Adding all these components yields a sum of just over
\$1,750,000, as shown in Table (5). A 10% contingency cost is added
to reach a total capital cost of \$1,925,000 for the total seawater
conditioning of the system.

As stated earlier, the air conditioning system for the conven-
tion complex will also include a complete conventional air condition-

system for reasons already discussed. The \$17.3M in capital costs in addition to the cost of that conventional system.

OPERATING COSTS

Although it is expected that the actual maintenance costs for the seawater air conditioning system will be less than for a conventional system, for this conservative analysis, it is assumed that the maintenance costs over the life of the system will be roughly the same as that for a conventional system. The largest difference is expected to come from the energy savings. These costs could be reduced between 70% and 90%.

ENERGY ANALYSIS

In general, one ton of cooling capacity requires about 1.0 kW of electricity, all of which would be derived from the electrical power used in the conventional air conditioning system. This energy is used to drive the circulation pumps, fans and compressors (or chillers). Chillers are the big energy consumers. In a typical air conditioning system, 800-900 kW of electricity is used per 1000 tons for the refrigeration alone. This is predicted to be replaced by 75-150 kW for 1000 tons of cooling for the seawater system (Van Ryzin, Leraand 1982).

The calculation done herein only includes the energy required to overcome the head loss due to friction in the cold water supply pipe. What is not considered is the head loss due to the resistance of the

t exchangers. Also, the friction loss of the effluent pipe would
o have to be considered. It may, therefore, appear reasonable to
ept the 75-100 kW per 1000 tons, since the projects described by
Ryzin and Leraand (1992) involved pumping the water to ultimate
tinations several meters above sea level, thereby increasing the
d loss.

Using an estimate of 100 kW per 1000 tons of capacity, the re-
t is 720 kW. The energy savings can then be computed as follows:

$$850 \text{ kW/1000 tons} - 100 \text{ kW/1000 tons} = 750 \text{ kW/1000 tons}$$

$$750 \text{ kW/1000 tons} \times 7.2 \text{ (1000 tons)} = 5400 \text{ kW}$$

This is the reduction in energy used when the system is running
full capacity. For the purpose of analysis, 4500 equivalent full
d hours is used for Honolulu. This is multiplied by the savings
full load to get the total annual energy savings in kilowatt-
rs,

$$5400 \text{ kW} \times 4500 = 24.3 \times 10^6 \text{ kWh}$$

Multiplying the annual savings in kilowatt-hours by the electric
e of about \$.10/kWh, the total annual savings to the convention
ter operator is calculated to be \$2.43 M.

CASH FLOW ANALYSIS

With an initial capital cost of \$17.3 M and an annual savings if
.3M, the payback period can be determined based on the net present
le of two options. The first option is constructing the seawater

conditioning system as described herein. The second option is the "nothing" option which would mean installing only the conventional conditioning system, and not realizing the annual energy savings not spending \$17.3 M in capital costs up front.

A rough evaluation by the Payback Method (White et al 1977) indicates that the payback period, based on a zero interest rate, would

$$\frac{\$17.3 \times 10^6}{2.43 \times 10^6 \$/\text{yr}} = 7.12 \text{ years}$$

An estimate of payback taking interest and inflation into account could assume an interest rate of 8% and an inflation rate of 3%. The combined effects could be assumed to be 5%. Using the compounding tables in the back of any economics book, a value for the number of periods of compounding can be applied. In this case, $n = 7.12$. In the "i = 5%" table, n is between 7 and 8 periods. Interpolation yields 7.55 years.

Both of the previous methods are fairly inexact. A more detailed analysis using 8% as the discount rate and 3% as a projected inflation rate can also be used. The analysis used by Syed et al (1991) used the following two formulas:

$$PW = \frac{A(1)}{(1+d)} \sum_{j=1}^{j=n} \left[\frac{(1+i)}{(1+d)} \right]^{j-1} \quad \text{Eq. (33)}$$

where:

PW = present worth of the revenue stream

(savings in this case)

i = inflation rate

d = discount rate

c = capital investment

A(j) = annual net revenue at the end of n years

= for any year j

n = the number of years (or periods of time)

When $d \neq i$, the geometric series can be written as,

$$S = A(1) \cdot \frac{\left[1 - \left[\frac{(1+i)}{(1+d)} \right]^n \right]}{(d-i)} \quad \text{Eq. (34)}$$

The payback period is the time N of value n at which S is equal to the capital cost, C. Therefore,

$$N = \frac{\log \left[\frac{1-C(d-i)}{A} \right]}{\log \left[\frac{(1+i)}{(1+d)} \right]} \quad \text{Eq. (35)}$$

In this particular case of the seawater air conditioning system,

$$N = \frac{\log \left[\frac{1-17.3(.08 - .03)}{2.43} \right]}{\log \left[\frac{(1+.03)}{(1+.08)} \right]} = 9.28 \text{ years}$$

This payback period is based on parameters such as the interest rate and inflation rate, which could easily change. Currently, the interest rate is closer to 7%. Using this in a similar calculation would yield a payback period of 8.80 years. Since the interest rates

the next several years appear to be more likely to rise a little, the more conservative estimate of a 9.28 year payback based on a 3% discount rate is favored here.

There are other variables as well. Not only is the capital investment estimate rough, but there is no experience in maintaining a seawater air conditioning system of this size, and therefore, the assumption that maintenance costs for the conventional versus seawater conditioning would be the same could also be a source of error.

A 9.28 year payback period may not be acceptable to the State, however, comparisons of payback periods, although a popular method, are not necessarily the best method of comparing the options. More responsible methods of comparing engineering options include comparing the present or future value of the resulting cash flows over the life expectancy.

The life expectancy is different for the different major components of the system. The supply pipeline is designed to last 50 years on the standpoint of strength. The heat exchangers, however, are expected to last only 10 to 20 years. An analysis taking these factors into account could compare the initial capital cost of \$17.3 M to the return of \$2.43 M per year, adjusted for inflation over a period of 20 years. In the pessimistic case, the heat exchangers could be replaced at a present cost of \$1.7 M at the 10 year mark. The comparison of present values would be calculated as follows. Assuming the interest rates increase at the 3% rate of inflation, the savings of

43 M per year in 1994 dollars over 20 years (a geometric series) equal to \$2.43 M (P/A, i, j, n), where $i = 8\%$, $j = 4\%$, $n = 20$ yrs. This is equal to $\$2.43 \text{ M} \times 61.7459 = \150 M . The present worth of the \$1.7 M heat exchangers replacement in 10 years is still \$1.7 M.

Therefore, the total present worth over a 20 year period is estimated to be:

- \$ 17.3 M initial capital costs
- +\$150.0 M present worth of electricity savings over 20 years
- \$ 1.7 M possible replacement of heat exchangers in 10 years

\$131.0 M

Therefore, the present worth of the seawater air conditioning system can be estimated to be \$131 M greater than the comparable conventional system.

Summary and Conclusions

Previous research has clearly shown that use of seawater air conditioning is technically feasible. The projects at Keahole Point and other locations have demonstrated that such a system can be designed, built and operated successfully.

The use of seawater air conditioning for Waikiki can be applied to the convention center project or perhaps other large building developments in Hawaii. The use of an HDPE pipeline of 1200 mm diameter would supply enough cold water to accomplish this.

The Ala Wai canal could provide a suitable conduit for a cold water supply pipeline in the case of the convention center, although a project of this magnitude could be expected to draw as much attention as the controversial convention center itself.

The problems of installing a pipeline through the nearshore area can be overcome with current technology and sound design procedures. Heat exchanger technology is also currently available "off the shelf," which could accomplish the desired goals of air conditioning for a convention center size project.

Conditions at Waikiki require a cold water supply pipe to extend at least 5600 meters offshore, to reach a minimum depth of 525 meters. The flow through a pipe of this length would not experience excessive temperature gains if the flow were fast enough and the pipe was of large enough diameter. This contributes to the argument that a seawater system is more feasible.

No adverse environmental effects are foreseen from the use of cold water, and fouling of either the cold water supply pipe or heat exchangers is also not expected to be a problem. The dredging operations off Waikiki could be an obstacle from the environmental standpoint, however, these types of operations are carried out frequently throughout Hawaii and the world, and procedures for minimizing environmental impact are standard.

To avoid much of the environmental impact, directional drilling or other tunneling techniques could be employed in the nearshore area, however, at a cost of \$500 to \$1000 per foot for a tunnel of the size being considered, the economics are prohibitive.

Overall, the use of seawater air conditioning for the convention center is estimated herein to cost approximately \$17.3 M more than a conventional system. Offsetting the cost is an estimated energy savings of \$2.43 M per year. This is calculated to provide a 7 to 9 year payback period, and a net savings of many tens of millions over the life of the system. This is the major incentive for constructing a system of this type.

The estimates and calculations used to arrive at these conclusions were preliminary and admittedly rough. As more building details become available, it is recommended each of the areas of the design be refined, and the cost benefits of a seawater air conditioning system for the Waikiki Convention Center be examined in more detail.

PIPE FLOW CALCULATIONS

Units:

$$1L \quad kg \equiv 1M \quad sec \equiv 1T \quad degC \equiv 1Q$$

$$= 1 \cdot kg \cdot \frac{m^2}{sec^3} \quad J := 1 \cdot kg \cdot \frac{m^2}{sec^2}$$

Parameters:

water density:

$$= 1027.6 \cdot \frac{kg}{m^3} \quad \text{For Seawater at 6 deg Celsius}$$

$$= 9.81 \cdot \frac{m}{sec^2}$$

water viscosity (kinematic):

$$= 1.56 \cdot 10^{-6} \cdot \frac{m^2}{sec} \quad \text{For Seawater at 6 deg Celsius}$$

pipe wall thickness is:

$$= .057 \cdot m$$

inside and outside pipe diameters are:

$$= 1.086 \cdot m \quad D_o := D + 2 \cdot t \quad D_o = 1.2 \text{ length}$$

total Pipe length:

$$= 7200 \cdot m$$

required Volumetric flow:

$$= 1.231 \cdot \frac{m^3}{sec}$$

ross sectional area of pipe (inside):

$$A = \pi \cdot \frac{D^2}{4} = 0.926 \text{ length}^2$$

ow velocity inside pipe:

$$V = \frac{Q}{A} = 1.329 \text{ length time}^{-1}$$

holds number for pipe flow:

$$Re_d = \frac{V \cdot D}{\nu} = 9.252 \cdot 10^5$$

ce the Reynolds Number is much grater than 2300, The flow is turbulent. In
s case a friction factor must be determined from empirical tables. Since a
yethelene pipe will be used, the smooth pipe assumption can be made.

m Appendix B-2 (Fox & McDonald, 1973) the friction factor is:

$$f = 0.0167$$

pressure change and head loss can then be calculated as follows:

$$\Delta P = \frac{V^2}{2} \cdot \left[f \cdot \frac{L}{D} \right] \cdot \rho = 1.005 \cdot 10^5 \text{ mass length}^{-1} \text{ time}^{-2}$$

l loss:

$$h_l = \frac{\Delta P}{\rho \cdot g} = 9.966 \text{ length}$$

Appendix A-2

gram to determine stresses on pipe due to hydrostatic pressure:

Material Characteristics:

Material: HDPE

Density: $\rho_p := 965 \text{ Kg/cu.meter}$

Poisson's Ratio: $\nu := .3$

Elongation modulus (50 yr):

$$E_{50} := \frac{200 \cdot 10^6}{.300} \text{ N/sq.meter}$$

Dimensions:

Length = 1.200 meters

$$\frac{OD}{2} = R_o = 0.6 \text{ meters}$$

Thickness = 21

$$\frac{OD}{SDR} = t = 0.057 \text{ meters}$$

Inner Diameter: $OD - 2 \cdot t$

$$ID = 1.086 \text{ meters}$$

$$\frac{ID}{2} = R_i = 0.543 \text{ meters}$$

Mass per unit length:

$$M = \rho_p \cdot \pi \cdot \left[R_o^2 - R_i^2 \right] \cdot 1 \quad \text{Mass} = 197.984 \text{ kg}$$

According to another criteria; the critical buckling strength can be calculated as follows:

Buckling Strength Coefficient: $C_B := .5$
 Correction factor for deflection: $C_D := 1.0$

Geometry dependent pipe stiffness:

$$PV_V := 53.7 \cdot E \cdot \frac{I}{ID^3}$$

Safety Factor: $FS := 2$

Then the critical buckling pressure is given by:

$$P_{cr} := C_B \cdot C_D \cdot \frac{\left[\frac{PV_V}{V} \right]^{.5}}{FS}$$

$$P_{cr} = 164.877 \text{ PSI}$$

SI units:

$$SIP_{cr} := P_{cr} \cdot 6895$$

$$SIP_{cr} = 1.137 \cdot 10^6 \text{ N/sq.meter}$$

oment of Inertia:

$$I = \frac{\pi \cdot t^3}{12} = 1.555 \cdot 10^{-5} \text{ meters}^4$$

Length: L := 7200 meters

ow Calculations:

Estimated A/C Load: LOAD := 7200 tons

Conversion factor for tons to Kilowatts: CONV := 3.5 KW/ton

Load In KW: LOADKW := LOAD · CONV

$$\text{LOADKW} = 2.52 \cdot 10^4 \text{ Kilowatts}$$

Specific Heat of Seawater: C_p := 3.985 KJ/Kg.deg C

Density of Seawater: ρ := 1027.6 Kg/cu.meter

Viscosity of Seawater: ν := 1.56 · 10⁻⁶ sq.meter/sec
(kinematic)

Expected Seawater temperature change in Heat Exchanger: δT := 5 degC

Required Mass flow:

$$M := \frac{\text{LOADKW}}{C_p \cdot \delta T} = 1.265 \cdot 10^3 \text{ Kg/sec}$$

Required Volumetric Flow:

$$Q := \frac{M}{\rho} = 1.231 \text{ cu.M/sec}$$

Converted to Gallons per minute:

$$Q_{\text{gal}} := Q \cdot 1000 \cdot .2642 \cdot 60 = 1.951 \cdot 10^4 \text{ Gal/min}$$

Flow Velocity:

$$V := \frac{Q}{\pi \cdot ID^2 \cdot .25} = 1.329 \text{ M/sec}$$

holds #:

$$Re := \frac{V \cdot ID}{\nu}$$

$$Re = 9.252 \cdot 10^5$$

ulation of Pressure loss, δP :

er Appendices B-1 and B-2, where f =friction factor:

$$f := .0167$$

$$\delta P := .5 \cdot V^2 \cdot f \cdot \frac{L}{ID} \cdot \rho$$

$$\delta P = 1.006 \cdot 10^5 \text{ N/sq.meter}$$

$$\delta P_{\text{psi}} := \frac{\delta P}{6895}$$

$$\delta P_{\text{psi}} = 14.585 \text{ psi}$$

Loss:

$$h_l := \frac{\delta P}{\rho \cdot g}$$

$$h_l = 97.863 \text{ meters}$$

ulate Collapse strength of pipe:

criteria for Hydrostatic collapse strength of round pipe is as follows:

$$p_c := \frac{24 \cdot E \cdot I}{50 \cdot (1 - \nu^2)} \cdot \frac{1}{\left[\frac{OD + ID}{2} \right]^3}$$

$$p_c = 1.832 \cdot 10^5 \text{ N/sq.Meter}$$

$$p_{\text{PSI}}_c = 26.563 \text{ PSI}$$

$$p_{\text{PSI}}_c := \frac{p_c}{6895}$$

Appendix A-3

CALCULATION OF TEMPERATURE CHANGE OF WATER IN PIPE

assumptions: 1. Flow over the outside of the pipe is constant.
2. C_p is constant for seawater in the pipe

define Variables:

pipe section length and average outside temperature:

$$L := 1200 \cdot \text{m} \quad T_o := 7.5 \cdot \text{degC}$$

$$f = 0.926 \cdot \text{length}^2$$

flow Temperature at the beginning of the pipe section:

$$T_i := 6.0 \cdot \text{degC}$$

$$n := 0.3 \quad n_2 := 0.4$$

flow velocity:

$$v = 1.329 \cdot \text{length}^{-1} \cdot \text{time}$$

$$k_p = 1.028 \cdot 10^3 \cdot \text{mass} \cdot \text{length}^{-3}$$

$$k_p := 1.51 \cdot \frac{W}{\text{m} \cdot \text{degC}}$$

$$1.56 \cdot 10^{-6} \cdot \text{length}^2 \cdot \text{time}^{-1}$$

Prandtl number:

$$Pr := 11.00$$

dynamic Viscosity of seawater:

$$\mu = \rho \cdot \nu = 0.002 \cdot \text{mass} \cdot \text{length}^{-1} \cdot \text{time}^{-1}$$

diameter based Reynolds Number for pipe flow:

$$Re = 9.252 \cdot 10^5$$

the temperature of the water outside the pipe is referred to as T_o . This changes as a function of depth as shown in fig (8) in the text and the depth also changes as a function of the length with respect of the pipe. Therefore the temperature is an unknown function of both depth and length of pipe. To calculate the heat loss over the length of the pipe, an average temperature outside the pipe could be used. A better solution can be arrived at by assuming an average temperature over various sections of the pipe and calculating the heat loss and temperature change for each section. The heat loss and temperature change in each section can then be added. This is the method employed.

The calculation for the one section is as follows:

for heating...

$$n := \frac{n}{2}$$

calculating the Nusselt number using Reynolds and Prandtl numbers for fully developed turbulent flow in smooth tubes:

$$Nu_d := 0.023 \cdot Re_d^{0.8} \cdot Pr_d^n \quad Nu_d = 3.558 \cdot 10^3$$

The heat transfer coefficient for the inside surface of the pipe is then calculated as follows:

$$h_i = \frac{k_p \cdot Nu_d}{D} = 4.948 \cdot 10^3 \text{ mass time}^{-3} \text{ temperature}^{-1}$$

The thermal resistance for the inside surface of the pipe is :

$$R_i = \frac{1}{h_i \cdot \pi \cdot D} = 5.924 \cdot 10^{-5} \text{ mass}^{-1} \text{ length}^{-1} \text{ time}^3 \text{ temperature}$$

For the unit length of polyethylene pipe the thermal resistance is calculated as follows, where the "r's" denote the inside and outside radii of the pipe and the k is the thermal conductivity of the pipe material.

$$R_p = \frac{D_o}{2k} = 1.51 \text{ mass length time}^{-3} \text{ temperature}^{-1}$$

$$\ln \left[\frac{r_o}{r_i} \right]$$

$$:= \frac{2 \cdot \pi \cdot k}{p}$$

$$R_p = 0.011 \text{ mass}^{-1} \text{ length}^{-1} \text{ time}^3 \text{ temperature}$$

For calculating the the heat transfer coefficient for the outside surface of the pipe, a flow velocity must be assumed and a Reynolds number calculated in order to choose an applicable empirical relationship.

Velocity of flow over the outer surface of the pipe:

$$:= 1.0 \cdot \frac{m}{\text{sec}}$$

Reynolds number for flow over outer surface of the pipe:

$$Re_f = \frac{D_o \cdot V_f}{\nu} = 7.692 \cdot 10^5$$

The Nusselt number is then:

$$Nu_f = 0.023 \cdot Re_f^{.8} \cdot Pr^{.4} = 3.07 \cdot 10^3$$

Therefore the heat transfer coefficient for the outside of the pipe can be estimated to be:

$$h_o = \frac{k_p \cdot Nu_f}{D} = 4.269 \cdot 10^3 \text{ mass}^{-3} \text{ time}^{-3} \text{ temperature}^{-1}$$

Based on the outer surface area of the pipe:

$$A_o = \pi \cdot D_o \cdot L = 4.524 \cdot 10^3 \text{ length}^2$$

$$R_o := \frac{1}{h_o \cdot A_o} \quad R_o = 5.179 \cdot 10^{-8} \text{ mass}^{-1} \text{ length}^{-2} \text{ time}^3 \text{ temperature}$$

The overall heat transfer coefficient for the pipe and seawater is :

$$U_o := \frac{1}{\frac{A_o}{A_i} \cdot \frac{1}{h_i} + \frac{A_o}{2 \cdot \pi \cdot k_p \cdot L_s} \ln \left[\frac{r_o}{r_i} \right] + \frac{1}{h_o}}$$

$$U_o = 0.974 \text{ mass}^{-3} \text{ time}^{-1} \text{ temperature}^{-1}$$

$$Q := U_o \cdot A_o \cdot \left[T_o - \left[T_{x1} + \frac{\delta T}{2} \right] \right] \quad \square$$

Calculation of the change in temperature within the pipe from the beginning of the section of interest to the end of that section.

The specific heat for seawater varies with temperature and salinity. For the purpose of this calculation the value from Appendix B-3 for 30% salinity and 0 deg Celsius is close enough.

Specific heat of seawater:

$$c_p := 4013.6 \cdot \frac{J}{kg \cdot \text{degC}}$$

Mass flow is calculated:

$$\dot{M} := \rho \cdot Q$$

$$\dot{M} = 1.265 \cdot 10^3 \text{ mass time}^{-1}$$

$$\dot{Q} = \dot{M} \cdot c_p \cdot \delta T \quad \square$$

to solve for the root, an initial estimate for δT is:

$$T := 0.1 \cdot \text{degC}$$

$$T := \text{root} \left[M \cdot C_p \cdot \delta T - U_o \cdot A_o \cdot \left[T_o - \left[T_{x1} + \frac{\delta T}{2} \right] \right], \delta T \right]$$

$$\delta T = 0.001 \text{ temperature}$$

Initial water temperature in the next section of pipe to be evaluated must be calculated by adding this temperature change.

$$T_{x2} := T_{x1} + \delta T \quad T_{x2} = 6.001 \text{ temperature}$$

This procedure is repeated for each section of pipe.

Pipe Section Number	Length (meters)	Avg. Depth (meters)	T ₀ (avg.) °C	T ₁ (entrance) °C	Δ T °C	T ₂ (end) °C
1	1200	450	7.5	6.000	8.67×10^{-4}	6.001
2	1800	325	11.0	6.001	.004	6.005
3	400	175	19.5	6.005	.011	6.016
4	800	60	26.0	6.016	.017	6.033
5	1200	10	26.5	6.033	.018	6.051
6	1200	2.5	27.0	6.051	.018	6.069
Totals	6600				.069	

Appendix A-4

following is a sample calculation designed to demonstrate how forces due offshore currents and incoming waves can affect a submerged pipeline. The parameters used are modeled after a pipeline sitting directly on a flat floor but they could be altered easily to represent other situations. The program is written in Mathcad version 2.5. This format allows the formulas to be easily read and explanations of each step inserted."

It is intended that the waves be transformed for shoaling, refraction and diffraction before use for actual design work. The Army Corps of Engineers Shore Protection Manual or published programs for that specific purpose would allow refraction and diffraction to be considered.

After determining the wave and current induced forces, the program executes an analysis of the free body diagram of a unit section of pipe. Parameters of pipe size and material can be altered to fit a given situation. The last part of the program predicts the amount of force required to stabilize the pipe under the given wave and current conditions. The assumption is made that the stabilizing forces would result from adding weight in the form of casting the pipe in concrete. The required force is translated into equivalent weight and volume of concrete.

For convenience, all the necessary parameters can be entered from the beginning of the program so that the results can be viewed on the same screen and scrolling through the program is required.

9 = the wave period of interest in seconds
 9.81 = gravity in metric units
 60 = local water depth in meters
 15 = wave height of interest in meters

The first step is to calculate the deep water wavelength, L_o

$$L_o := \frac{g}{2\pi} T^2$$

$$L_o = 126.466 \text{ meters}$$

An iterative process is required to calculate the local wavelength. To accomplish this the program requires an initial estimate.

$L_o = 30$ can be used as an initial estimate.

The calculation of wavelength is as follows:

$$L := \text{root} \left[\frac{d}{L} - \frac{d}{L} \tanh \left[2\pi \frac{d}{L} \right], L \right]$$

$$L = 125.826 \text{ meters}$$

The wave number is calculated to simplify the calculation of the wave heights.

$$k := 2\pi \frac{1}{L}$$

The calculation of the local wave height resulting from the deep water wave incident on the local region could then be calculated for an in-depth analysis of wave conditions in a given area. However, for the purpose of this calculation, the local wave height is assumed to be H meters.

$$H = 15 \text{ meters}$$

following values can be computed and compared to those found in table C-1 of the Shore Protection Manual.

$$\frac{d}{L_o} = 0.474$$

$$\frac{d}{L} = 0.477$$

test the local wave height for breaking.

$$H := \text{if} \left[\frac{H}{d} > 0.78, H, 0 \right]$$

local) is H when $H/d > .78$, otherwise assume it is 0

Summary of Wave Information Results:

$L = 15$ meters $T = 9$ seconds $d = 60$ meters

$$\frac{H}{d} = 0.25$$

ANALYSIS OF PARTICLE MOTIONS:

Calculation of water particle velocities and accelerations is as follows

Let $\omega := 2 \frac{\pi}{T}$ and Ω is the phase angle of the wave, evaluated at the

point. $z := -d$ Several values of Ω are examined so that the maximum

values can be determined. Since the 'worst case' expected is for the

direction of wave propagation to be at a 60 degree angle of incidence, α on

the pipe, the velocities and accelerations in the following equations include

a factor $\cos \alpha$.

1.047 = radians angle of incidence

0.15 .. 180 = phase angle in degrees

$= j \frac{\pi}{180}$ = phase angle in radians

$$= \frac{H g k}{2 \omega} \left[\frac{\cosh(k(d+z))}{\cosh(kd)} \right] \sin \left[\frac{\Omega}{j} \right] \sin(\alpha)$$

dt=a

$$= \frac{-(H g k) \cosh(k(d+z))}{2 \cosh(kd)} \cos \left[\frac{\Omega}{j} \right] \sin(\alpha)$$

j	degrees
0	
15	
30	
45	
60	
75	
90	
105	
120	
135	
150	
165	
180	

CULATION OF WAVE INDUCED FORCES:

calculate the forces due to the particle velocities resulting from wave
 on the following equations are employed. First however, the coefficients
 drag, lift and added mass must be assumed.

$$C_D = 0.5 \quad C_L = 1.4 \quad C_m = 3.3$$

er characteristics of the pipe geometry and fluid density are also
 required.

= 1.1 meters = approximate outside diameter

1.027 kg/cu.meter = density of seawater

equation for the horrizontal force, Fx is the Morrison's Equation.

$$F_{xj} = C_m \rho \pi \left[\frac{D}{2} \right]^2 a_j + \frac{1}{2} \rho D C_D C_d u_j^2$$

the verticle direction per unit length, the following equation is used.

$$F_z = C \frac{\rho}{2} D C u_j^2$$

F_x j Newtons	F_z j Newtons
-1.022 10	10.94
-983.097	40.828
-870.342	81.656
-693.374	122.484
-467.167	152.372
-210.048	163.312
58.326	152.372
318.885	122.484
554.655	81.656
751.7	40.828
899.505	10.94
990.911	0
1.022 10	

FORCES RESULTING FROM STEADY CURRENT: (Alone)

There is also a drag force and lift force associated with the steady current incident on the pipeline. The same equations are used to calculate the forces in the x and z-directions with the velocity of the current used instead of particle velocities. Note that there is no acceleration since the current is 'steady' and therefore the inertial portion of the Morrison's equation is equal to zero.

A 1 knot current, $U = 1\text{m/sec}$, is assumed to be present in the longshore direction. This is assumed to be in a perpendicular direction to the direction of the pipe.

$U = 1 \text{ m/s}$ = steady current velocity

$$F_{x \text{ current}} = \frac{1}{2} \rho D C_d U^2$$

$$F_{z \text{ current}} := C_L \frac{\rho}{2} D_o U^2$$

$$F_{z \text{ current}} = 282.425 \text{ NEWTONS} \quad F_{z \text{ current}} = 790.79 \text{ NEWTONS}$$

ing the components for the x and z-directions for both the wave
 ced motion and the current velocities doesn't necessarily yield a total
 e of the water on the pipeline per unit area. To examine the total
 ultant forces on the pipeline for a given section under given wave
 ditions, the partial velocities and accelerations from the waves and
 rents must be added to get an effective velocity, U_e and acceleration, a_e .
 the combined particle motions can be used in the Morrison's and lift
 ation.

$$U_e := U + u_j$$

$$a_e := a_j \quad \text{Since the current is assumed to be steady.}$$

indices on the wave forces remain from the previous calculations
 use the phase angle at which the 'worst' case for the total resultant
 es on the pipe is not yet known.

$$F_{x_j} := C_m \rho \pi \left[\frac{D_o}{2} \right]^2 a_{e_j} + \frac{1}{2} \rho D_o C_d U_{e_j}^2$$

$$F_{z_j} := C_L \frac{\rho}{2} D_o U_{e_j}^2$$

ANALYSIS OF ALL FORCES ON THE PIPELINE SECTION

the total forces at various wave phase angles determined, only the forces the weight of the pipe, bouyancy of the pipe and friction with the bottom can to be determined for an analysis to be complete. The bouyancy and weight of the pipe are constant, however the friction force resisting movement in the x-direction is dependant on the normal force of the pipe on bottom. The normal force is the sum of all the vertical forces and varies with the wave phase angle.

Bouyant, F_b , force per unit length on the pipeline is calculated as follows.

$$F_b := \rho \cdot g \cdot \pi \cdot \frac{D_o^2}{4}$$

where: ρ = density of seawater
 D_o = pipe diameter (outside)

$$F_b = 9.574 \cdot 10^3 \text{ Newtons/unit length}$$

Weight of the pipe per unit length is the sum of the weight of the pipe material itself plus the weight of the water contained. The density of the material, ρ_p , and the inside diameter, D_i , of the pipe are required for calculation.

1 meter $\rho_p = 954 \text{ kg/cu. meter}$

$$W_p := \rho_p \cdot g \cdot \pi \cdot \frac{D_i^2}{4} + \rho \cdot g \cdot \pi \cdot \frac{D_o^2 - D_i^2}{4}$$

3

$$W = 9.456 \cdot 10^3 \text{ Newtons/unit length}$$

inspection, it can be seen that added weight is required in order to overcome the bouyant force so that the pipe will stay on the seafloor. To determine the downward force per unit length required to accomplish this, the sum of all the verticle forces, F_z , must be determined.

$$F_{zTj} := F_{zj} + F_b - W_p$$

F_{zTj}	Newton
908.904	
1.106	10^3
1.309	10^3
1.499	10^3
1.654	10^3
1.756	10^3
1.791	10^3
1.756	10^3
1.654	10^3
1.499	10^3
1.309	10^3
1.106	10^3
908.904	

force required to overcome bouyancy throughout the wave cycle is the maximum value of F_z .

$$F_z := \max[F_{zT}]$$

$$F_z = 1.791 \cdot 10^3 \text{ Newtons/unit length}$$

volume concrete, V_c , required to weight the pipe is calculated as follows.

weight of the required concrete in seawater water is equal to F_z .

$$F_z$$

$$= 2.4 \times 10^3 \text{ Kg/cu. meter}$$

crete

$$V_c := \frac{W_c}{[\rho_{\text{concrete}} - \rho] g}$$

$$= 0.133 \text{ cubic meters of concrete/meter length of pipe}$$

amount of concrete is the minimum required to overcome the bouyant and
 t case lift force for water motion. This may not be enough to provide the
 al force on the seafloor required for friction to overcome the maximum
 force in the x-direction. To accomplish this, the weight of the concrete
 ired is calculated for each phase angle and the maximum is taken. This is
 because the the maximum lift affects induces a minimum normal force but
 will not occur at the phase angle at which the maximum drag occurs.
 sum of the forces in the x-direction, F_{xT} , is calculated using an assumed
 efficient of friction, f , of 0.5 applied to the normal forces.

0.5

$$F_{xTj} := F_{xj} - f [F_{zTj} - W_c]$$

maximum of the sum computed above is the resultant force which must be
 rome by adding wieght to induce a greater normal force. The amount of
 ion normal force required is equal to the resultant maximum resultant
 , F_x , devided by the coefficient of friction. Again this normal force
 e applied by adding concrete weights, W_{ac} , to the pipe.

$$W_{ac} := \frac{\max \left[\begin{matrix} F_x \\ F_z \end{matrix} \right]}{f}$$

$$W_{ac} = 3.491 \cdot 10^3$$

total downward force required by concrete attached to the pipe is the sum of the concrete added to overcome the lift and bouyancy and that added to overcome drag.

$$:= W_{ac} + W_c$$

$$= 5.281 \cdot 10^3$$

volume of concrete is computed in a similar manner as previously.

$$V_T := \frac{F_T}{\left[\rho_{\text{concrete}} - \rho \right] g}$$

$$V_T = 0.392 \quad \text{cubic meters per unit length}$$

and, this amount of concrete weighs:

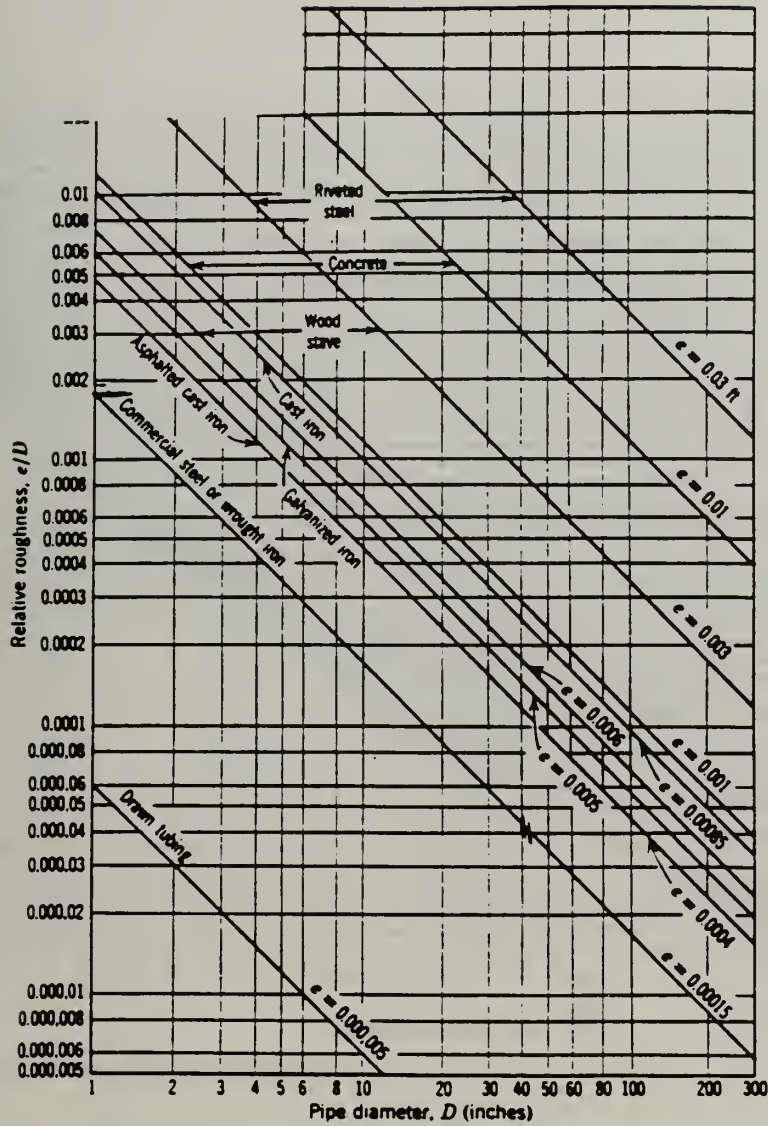
$$W_T := V_T \rho_{\text{concrete}} g$$

$$W_T = 9.232 \cdot 10^3 \quad \text{Newtons}$$

Sample Results

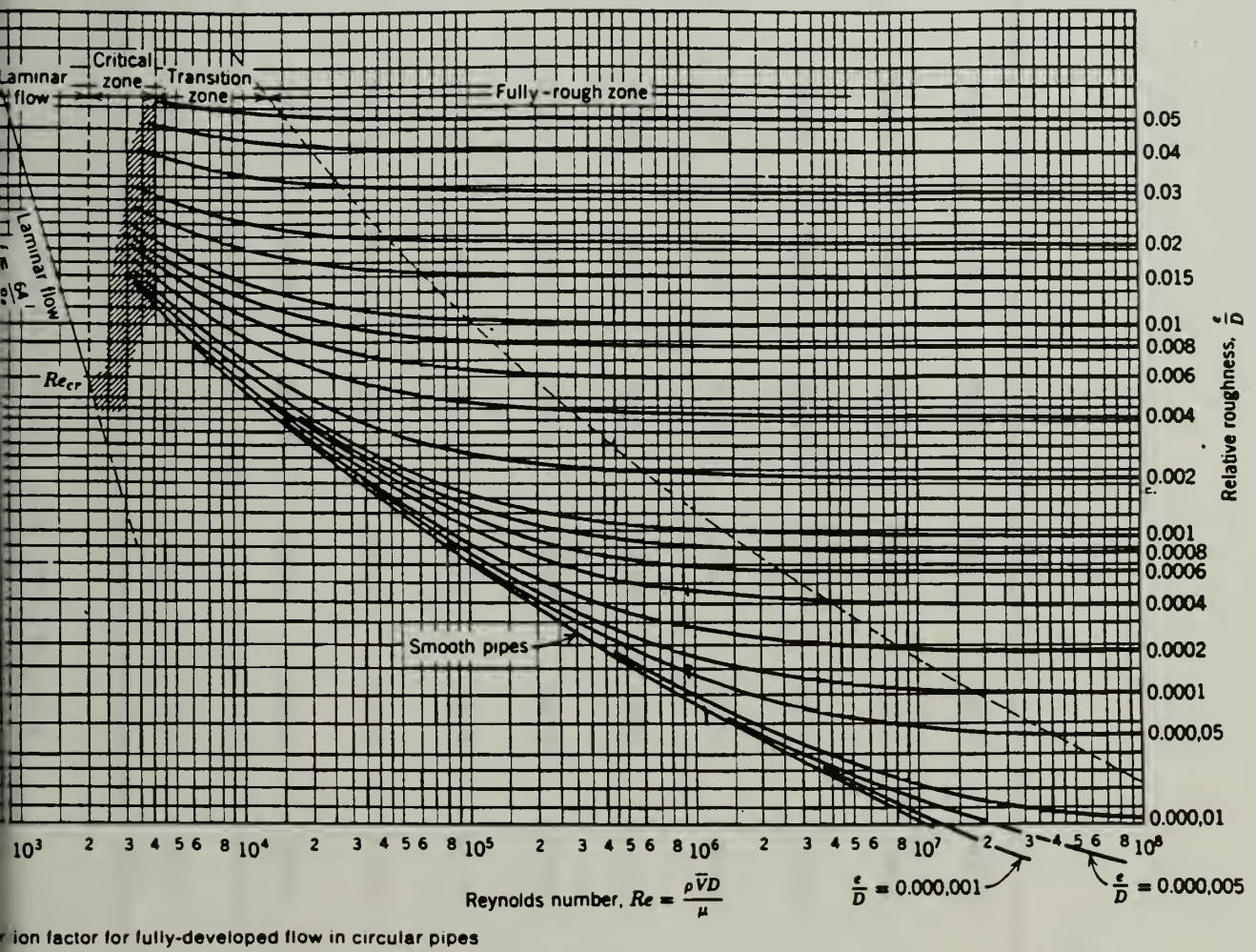
Water Depth (meters)	Wave Height, H _o (meters)	Period, T (seconds)	Min. Vertical Applied Force Required per Meter Length of Pipe for Stability (Newtons)
200	9	9	1476
200	9	10	1485
200	9	12	1574
200	9	15	2016
200	15	9	1477
200	15	10	1492
200	15	12	1641
200	15	15	2392

Appendix B-1



Relative roughness values for pipes of common engineering materials

Appendix B-2



Specific Heat of Seawater at Constant Pressure ($Jg^{-1}C^{-1}$) at Various Salinities and Temperatures

Salinity, 0/00	0°C	5°C	10°C	15°C	20°C	25°C	30°C	35°C	40°C
0	4.2174	4.2019	4.1919	4.1855	4.1816	4.1793	4.1782	4.1779	4.1783
5	4.1812	4.1679	4.1599	4.1553	4.1526	4.1513	4.1510	4.1511	4.1515
10	4.1466	4.1354	4.1292	4.1263	4.1247	4.1242	4.1248	4.1252	4.1256
15	4.1130	4.1038	4.0994	4.0982	4.0975	4.0977	4.0992	4.0999	4.1003
20	4.0804	4.0730	4.0702	4.0706	4.0709	4.0717	4.0740	4.0751	4.0754
25	4.0484	4.0428	4.0417	4.0437	4.0448	4.0462	4.0494	4.0508	4.0509
30	4.0172	4.0132	4.0136	4.0172	4.0190	4.0210	4.0251	4.0268	4.0268
35	3.9865	3.9842	3.9861	3.9912	3.9937	3.9962	4.0011	4.0031	4.0030
40	3.9564	3.9556	3.9590	3.9655	3.9688	3.9718	3.9775	3.9797	3.9795

From Riley and Skirrow, 1975

mechanical, Thermal and Processing Properties of Glass Fiber Reinforced Plastics

Material	Specific Gravity	Thermal Conductivity W/(m K)	Specific Heat, J/(kg K) ^b	Tensile Strength MPa	Tensile Modulus, GPa	Flexural Modulus, GPa	Compressive Strength, MPa	Impact Strength Load at 22°C, J/m
Glass fiber-reinforced thermoplastics (RTP)	1.04	14.5		45	3.7	3.6	172	59
poly(phenylene sulfide)	1.64	3.47	1.05	152	14.1	13.1	145	80
acrylonitrile-butadiene-styrene terpolymer (ABS)	1.22	2.42		76	6.2	6.0	97	64
poly(phenylene oxide) (PPO)	1.21	6.57	.84 - 1.67	100	6.3	5.2	121	96
poly(ethylene terephthalate)	1.56	11.2		145	9.0	8.6	172	96
Unreinforced thermoplastics (TP)								
poly carbonate	1.20	2.34	1.26	66	2.3	2.3	86	854
polypropylene	0.89	2.10	1.88	34	0.7	0.9-1.4	24	50-1000
poly(phenylene sulfide)	1.30	2.89		66	3.3	3.8	110	<27

styrene
terpolymer
(ABS)

poly(phenylene
oxide) (PPO) 1.10 1.59 0.84-1.67 54 2.6 2.3-2.8 83 270

styrene-
acrylonitrile
(SAN) 1.05 1.21 1.38 66 2.8 3.8 97 16-24

poly(butylene
terephthalate) 1.31 1.76-2.89 57 1.9 2.3-2.8 59 43

poly(ethylene
terephthalate) 1.34 1.51 1.42 59 2.8 2.4-3.1 76 13-35

ASTM A-606
HSLA steel,
cold rolled 7.75 43.3 0.46 448 207 448

SAE 1008 low
carbon steel,
cold rolled 7.86 60.6 0.42 331 207 331

Metals
AISI 304
stainless steel 8.03 16.3 0.50 552 193 552

TA 2036
aluminum,
wrought 2.74 159 0.88 338 70 338

ASTM B85
aluminum, die
cast 2.82 91.8 71 331 331

ASTM A40A
zinc, die cast

6.59

113

0.42

283

75

283

From Rosato and Rosato, 1990

Material	Modulus of elasticity E, GN / m ²	Poisson's ratio, ν	Shear modulus G, GN / m ²	Coefficient of linear expansion α , 10 ⁻⁶ / °C	Mass Density 10 ³ kg / m ³
Aluminum	68-78.6	0.32-0.34	25.5-26.5	20.0-24.1	2.66-2.88
Iron, cast	89-145	0.21-0.30	35.8-56.5	10.4	6.95-7.34
Steel	193-220	0.26-0.29	75.8-82.0	9.9-12.8	7.72-7.86
Stainless Steel	193-207	0.30	73.1	14.9-16.9	7.64-7.92
Titanium	106-114	0.34	41.4	8.82	4.51
Glass	50-79	0.21-0.27	26.2-32.4	5.94-9.54	2.38-3.88
Polyethylene	0.14-0.38	0.45	0.117	180	0.91
Rubber	0.00076-0.0041	0.50	0.0003-0.001	126-198	1.0-1.24

From Crandall et al 1978

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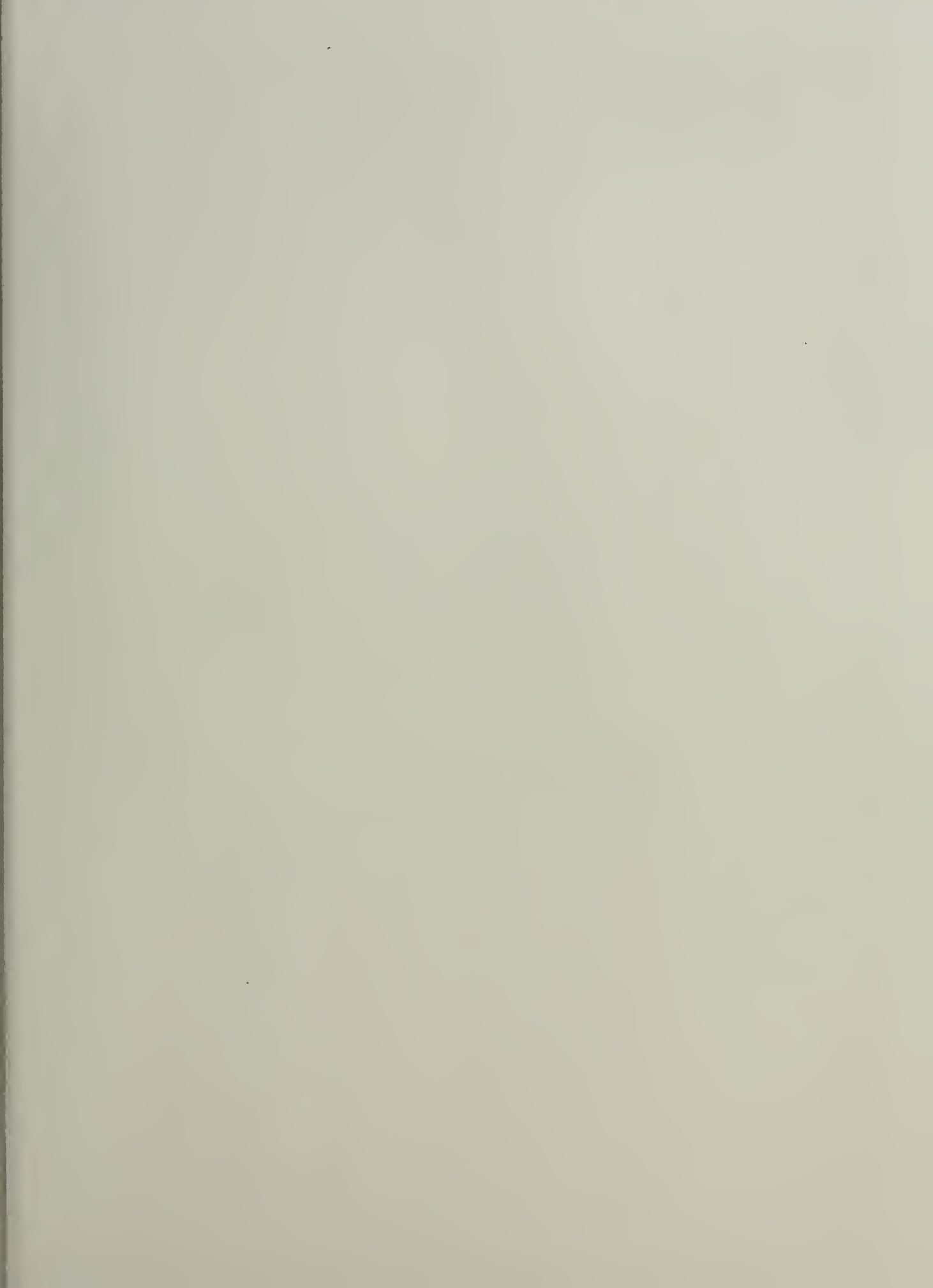
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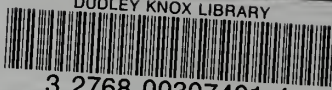


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